Heating · Piping and Air Conditioning

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THE EVAPORATIVE CONDENSER

Theory and Characteristics

THE evaporative condenser combines in one piece of apparatus two of the oldest elements in refrigeration: the condenser and the cooling tower. Unlike a separate condenser and cooling tower, however, the condensing coil of the evaporative condenser is directly in the spray of water, and the air is drawn through both the condensing coil and the spray water as shown in Fig. 1.

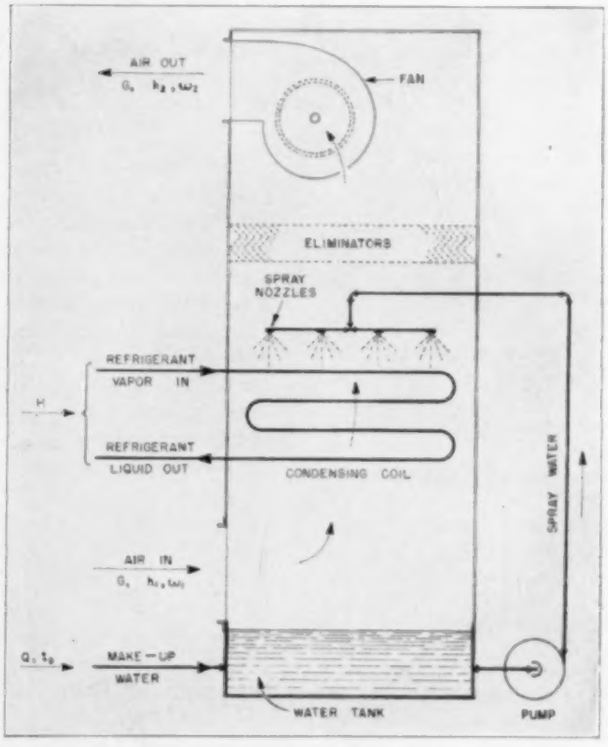

Fig. 1—Schematic diagram of an evaporative condenser, showing the symbols used in the accompanying text

The refrigerant is condensed inside of the coil. Water is sprayed continuously over the outside of the coil. The spray water is collected in a tank at the bottom of the condenser. A pump withdraws the water from the tank and delivers it to the spray nozzles. The same water is circulated continuously, and is neither heated nor cooled during its passage through the pipes connecting the tank, pump, and spray nozzles. Inasmuch as the spray water wets the outside surface of the coil, the heat is transferred through the wall of the coil to the water on its outer surface. But, the water as fast as it receives this heat transfers it in turn to the air flowing over the coil. As the water is neither heated nor cooled while it is circulated, it must attain an equilibrium temperature that remains constant as long as the operating conditions remain unchanged.

The equilibrium temperature of the spray water must always be above both the initial and final wet bulb temperatures of the air or heat could not be transferred from the water to the air. Therefore, the temperature of the water will also be above the dew point temperature of the air at all points in the coil. Because of this, and also because of the heat supplied to the water by the condensing refrigerant, evaporation of water takes place into the air. As the air flows through the condenser, its absolute humidity (moisture content) is increased.

The temperature of the spray water may be either above or below the initial dry bulb temperature of the air. In either case, there will be a transfer of sensible heat between the air and water in addition to a transfer of moisture. As a result, both the dry bulb and dew point temperatures of the air will change. The total heat surrendered by the condensing refrigerant is ultimately transferred to the air and results in an increase in its enthalpy (total heat content).

By William Goodman*

The evaporative condenser is an item of air conditioning and heat transfer equipment that has developed rapidly in recent months and is finding ever increasing usage. Very little authoritative engineering information about this type of apparatus has been generally available until now. . . . In this discussion, the author gives basic methods and equations, with examples showing their use, describes how to compute heat transferred by evaporative condensers, rating of a condenser, selection of air quantity and coil area, and the spray water temperature and theoretical maximum capacity of condensers. His tables and curves have been designed to simplify the solution of evaporative condenser problems

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Part 1.

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Symbols

A = area of outside surface of condensing coil, sq ft (area in contact with spray water).
B = ratio of outside to inside surface of coil.
$\frac{A}{B}$ = area of inside surface of condensing coil, sq ft (area in contact with condensing refrigerant).
f_a = coefficient of heat transfer between air and water (air film coefficient), Btu per hr per sq ft of surface per degree Fahrenheit difference.
f_R = coefficient of heat transmission between condensing refrigerant and metal tube wall (refrigerant film coefficient), Btu per hr per sq ft per F.
f_w = coefficient of heat transmission between water and metal wall (water film coefficient), Btu per hr per sq ft per F.
G = weight of air circulated through the condenser, lb per hr.
H = total heat surrendered by the condensing refrigerant—that is, the total heat capacity of the condenser, Btu per hr.
h_1 = enthalpy (heat content) of air entering condenser, Btu per lb.
h_2 = enthalpy of air leaving condenser, Btu per lb.
h_R = enthalpy of air at a wet bulb temperature equal to the temperature of the condensing refrigerant, Btu per lb.
h_w = enthalpy of air at a wet bulb temperature equal to the temperature of the spray water, Btu per lb.
M = a quantity defined by equation 2.
N = a quantity defined by equation 3.
N_L = limiting value of N as defined by equation 6.
t_1 = initial dry bulb temperature of the air, F.
t_2 = final dry bulb temperature of air leaving condenser, F.
t_1' = initial wet bulb temperature of the air, F.
t_2' = final wet bulb temperature of air leaving condenser, F.
t_R = temperature of condensing refrigerant, F.
t_w = temperature of spray water, F.
U = coefficient of heat transmission defined by equation 5, Btu per hr per sq ft per F.
w = absolute humidity of air, lb per lb of dry air.
Z = a quantity that depends on M; see Table 1.

How to Compute Total Heat Transferred

There is no need for separately evaluating the transfer of sensible heat and moisture between the air and water. As shown in appendix 1, the total heat transferred from the water to the air depends only upon the wet bulb temperature of the air, and can be computed by means of the following equation:

$$H = G\,(h_w - h_1)\,Z \quad \text{.....[1]}$$

The factor Z can be found in Table 1, after first computing another factor M, by means of the following formula:

$$M = \frac{f_w A}{0.24G} \quad \text{.....[2]}$$

After computing M for a given set of conditions, the factor Z may be found in Table 1. The total quantity of heat that can be transferred from the water wetted surface of the coil to the air in contact with it can be computed by means of equation 1.

Before equation 1 can be used, the spray water temperature must be known in order to find h_w. If, as is commonly the case, the condensing temperature of the refrigerant and the initial wet bulb temperature of the air are known, the spray water temperature can be computed. For this purpose, Fig. 2, Table 2, and the following equation are used. (See appendix 2 for derivation.)

$$N = \frac{GZ}{AU} \quad \text{.....[3]}$$

After the spray water temperature has been computed, the total refrigerating capacity of the condenser can be computed by means of either equation 1 or 4.

$$H = U\,(t_R - t_w)\,A \quad \text{.....[4]}$$

(see appendix 2)

U is the overall coefficient of heat transmission from the condensing refrigerant to the spray water. It does *not* include the film coefficient from the air to the water. As shown in appendix 2, the value of U is computed by means of:

$$\frac{1}{U} = \frac{1}{f_w} + \frac{B}{f_R} \quad \text{.....[5]}$$

The method of using Fig. 2 for finding the spray water temperature is illustrated by Fig. 3. After N has been computed for a particular condenser, the corresponding angle θ can be found in Table 2. Then point A of Fig. 3 is located at the intersection of the horizontal line representing the *initial wet bulb temperature* of the air and the vertical line representing the *temperature of the condensing refrigerant*. Through point A draw a straight line at an angle θ with the vertical line. This line intersects the curve at B, at which point the temperature of the spray water is read.

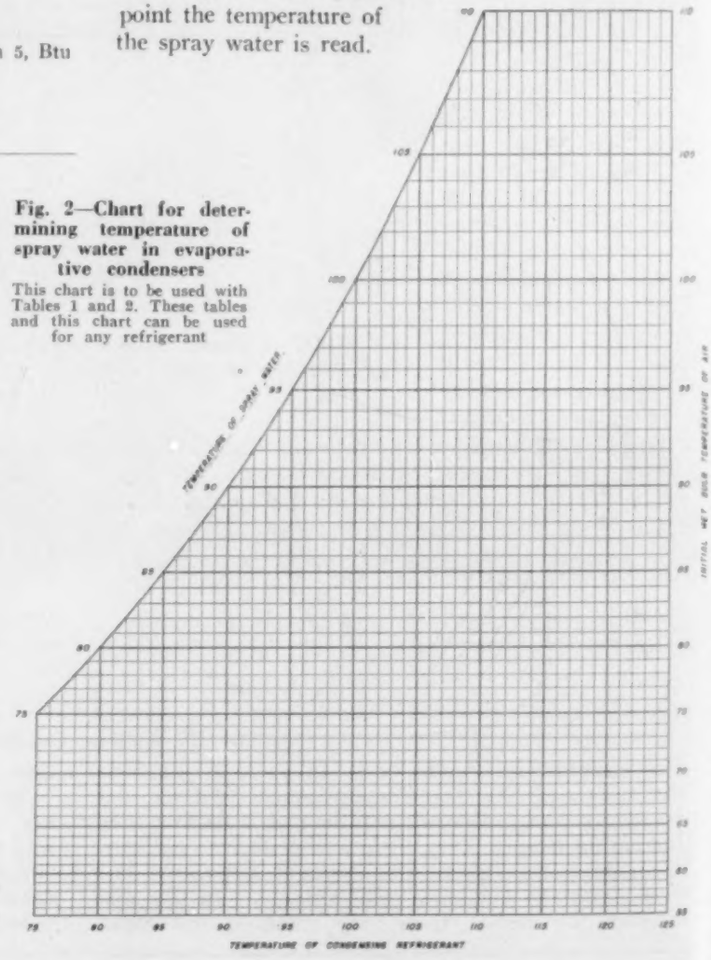


Fig. 2—Chart for determining temperature of spray water in evaporative condensers
This chart is to be used with Tables 1 and 2. These tables and this chart can be used for any refrigerant

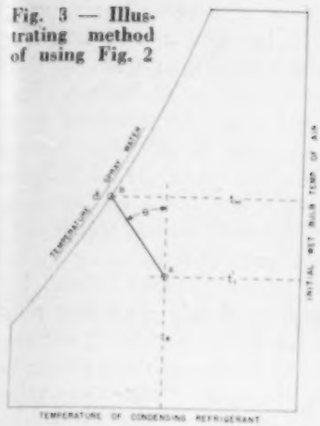


Fig. 3 — Illustrating method of using Fig. 2

Example 1: Find the capacity of a "Freon" evaporative condenser if the head pressure is to be maintained at 121 lb gage and the wet bulb temperature of the available air is 75 F. The condenser fan delivers 2500 cfm. The condenser coil has 500 sq ft of outside area and B equals 10. The values of the various film coefficients are, $f_a = 10$, $f_w = 75$, and $f_R = 250$.

Solution: At a pressure of 121 lb gage, the condensing temperature of "Freon" is 102 F.

$$\frac{1}{U} = \frac{1}{f_w} + \frac{B}{f_R} \quad \text{.....[5]}$$

$$= \frac{1}{75} + \frac{10}{250} = 0.0533$$

$$U = 18.8$$

Assuming the density of air to be 0.075 lb per cu ft, weight of air delivered by the fan is:

$$G = 2500 \times 60 \times 0.075$$

$$= 11{,}250 \text{ lb per hr}$$

$$M = \frac{f_a A}{0.24G} \quad \text{.....[2]}$$

$$= \frac{10 \times 500}{0.24 \times 11{,}250}$$

$$= 1.85$$

Referring to Table 1, for $M = 1.85$, $Z = 0.8428$.

$$N = \frac{GZ}{AU} \quad \text{.....[3]}$$

$$= \frac{11{,}250 \times 0.8428}{18.8 \times 500}$$

$$= 1.01$$

Referring to Table 2, for $N = 1.01$, $\theta = 45°\text{-}17'$.
Using Fig. 2, for $t_R = 102$ F and $t_1' = 75$ F, $t_w = 87.7$ F.
The capacity of the condenser can be computed by either equations 1 or 4. As a check, the capacity will be computed by both.

$$H = G\,(h_w - h_1)\,Z \quad \text{.....[1]}$$

Interpolating in the psychrometric tables,[1] for $t_w = 87.7$ F, $h_w = 52.73$ Btu, and for $t_1' = 75$ F, $h_1 = 38.53$

$$H = 11{,}250\,(52.73 - 38.53)\,0.8428$$
$$= 11{,}250 \times 14.20 \times 0.8428$$
$$= 134{,}600 \text{ Btu per hr}$$

$$H = U\,(t_R - t_w)\,A \quad \text{.....[4]}$$
$$= 18.8\,(102 - 87.7)\,500$$
$$= 18.8 \times 14.3 \times 500$$
$$= 134{,}400 \text{ Btu per hr}$$

Table 1—Values of Factors M and Z

M	Z	M	Z	M	Z
0.00	0.0000	0.45	0.3624	1.30	0.7275
0.01	0.0100	0.46	0.3687	1.32	0.7329
0.02	0.0198	0.47	0.3750	1.34	0.7381
0.03	0.0296	0.48	0.3812	1.36	0.7433
0.04	0.0392	0.49	0.3874	1.38	0.7484
0.05	0.0488	0.50	0.3935	1.40	0.7534
0.06	0.0582	0.52	0.4055	1.42	0.7583
0.07	0.0676	0.54	0.4173	1.44	0.7631
0.08	0.0769	0.56	0.4288	1.46	0.7678
0.09	0.0861	0.58	0.4401	1.48	0.7724
0.10	0.0952	0.60	0.4512	1.50	0.7769
0.11	0.1042	0.62	0.4621	1.55	0.7878
0.12	0.1131	0.64	0.4727	1.60	0.7981
0.13	0.1219	0.66	0.4831	1.65	0.8080
0.14	0.1306	0.68	0.4934	1.70	0.8173
0.15	0.1393	0.70	0.5034	1.75	0.8262
0.16	0.1479	0.72	0.5132	1.80	0.8347
0.17	0.1563	0.74	0.5229	1.85	0.8428
0.18	0.1647	0.76	0.5323	1.90	0.8504
0.19	0.1730	0.78	0.5416	1.95	0.8577
0.20	0.1813	0.80	0.5507	2.00	0.8647
0.21	0.1894	0.82	0.5596	2.10	0.8775
0.22	0.1975	0.84	0.5683	2.20	0.8892
0.23	0.2055	0.86	0.5768	2.30	0.8997
0.24	0.2134	0.88	0.5852	2.40	0.9093
0.25	0.2212	0.90	0.5934	2.50	0.9179
0.26	0.2289	0.92	0.6015	2.60	0.9257
0.27	0.2366	0.94	0.6094	2.70	0.9328
0.28	0.2442	0.96	0.6171	2.80	0.9392
0.29	0.2517	0.98	0.6247	2.90	0.9450
0.30	0.2592	1.00	0.6321	3.00	0.9502
0.31	0.2666	1.02	0.6394	3.10	0.9550
0.32	0.2739	1.04	0.6465	3.20	0.9592
0.33	0.2811	1.06	0.6535	3.30	0.9631
0.34	0.2882	1.08	0.6604	3.40	0.9666
0.35	0.2953	1.10	0.6671	3.60	0.9727
0.36	0.3023	1.12	0.6737	3.80	0.9776
0.37	0.3093	1.14	0.6802	Inf.	1.0000
0.38	0.3161	1.16	0.6865		
0.39	0.3229	1.18	0.6927		
0.40	0.3297	1.20	0.6988		
0.41	0.3364	1.22	0.7048		
0.42	0.3430	1.24	0.7106		
0.43	0.3495	1.26	0.7163		
0.44	0.3560	1.28	0.7220		

Table 2—Angles for Fig. 2

N	θ	N	θ	N	θ	N	θ
0.00	0°— 0′	0.50	26°—34′	1.40	54°—28′	5.00	78°—41′
0.02	1°— 9′	0.52	27°—28′	1.42	54°—51′	5.20	79°— 7′
0.04	2°—17′	0.54	28°—22′	1.44	55°—13′	5.40	79°—31′
0.06	3°—26′	0.56	29°—15′	1.46	55°—35′	5.60	79°—53′
0.08	4°—34′	0.58	30°— 7′	1.48	55°—57′	5.80	80°—13′
0.10	5°—43′	0.60	30°—58′	1.50	56°—19′	6.00	80°—32′
0.11	6°—17′	0.62	31°—48′	1.55	57°—10′	6.20	80°—50′
0.12	6°—51′	0.64	32°—37′	1.60	58°— 0′	6.40	81°— 7′
0.13	7°—24′	0.66	33°—25′	1.65	58°—47′	6.60	81°—23′
0.14	7°—58′	0.68	34°—13′	1.70	59°—32′	6.80	81°—38′
0.15	8°—32′	0.70	35°— 0′	1.75	60°—15′	7.00	81°—52′
0.16	9°— 6′	0.72	35°—45′	1.80	60°—57′	7.20	82°— 6′
0.17	9°—39′	0.74	36°—30′	1.85	61°—36′	7.40	82°—18′
0.18	10°—12′	0.76	37°—14′	1.90	62°—14′	7.60	82°—30′
0.19	10°—45′	0.78	37°—57′	1.95	62°—51′	7.80	82°—42′
0.20	11°—19′	0.80	38°—40′	2.00	63°—26′	8.00	82°—52′
0.21	11°—52′	0.82	39°—21′	2.10	64°—32′	8.20	83°— 3′
0.22	12°—24′	0.84	40°— 2′	2.20	65°—33′	8.40	83°—13′
0.23	12°—57′	0.86	40°—42′	2.30	66°—30′	8.60	83°—22′
0.24	13°—30′	0.88	41°—21′	2.40	67°—23′	8.80	83°—31′
0.25	14°— 2′	0.90	41°—59′	2.50	68°—12′	9.00	83°—40′
0.26	14°—34′	0.92	42°—37′	2.60	68°—58′	9.20	83°—48′
0.27	15°— 7′	0.94	43°—14′	2.70	69°—41′	9.40	83°—56′
0.28	15°—39′	0.96	43°—50′	2.80	70°—21′	9.60	84°— 3′
0.29	16°—10′	0.98	44°—25′	2.90	70°—58′	9.80	84°—10′
0.30	16°—42′	1.00	45°— 0′	3.00	71°—34′	10.00	84°—17′
0.31	17°—13′	1.02	45°—34′	3.10	72°— 7′	11.00	84°—48′
0.32	17°—45′	1.04	46°— 7′	3.20	72°—39′	12.00	85°—14′
0.33	18°—16′	1.06	46°—40′	3.30	73°— 8′	13.00	85°—36′
0.34	18°—47′	1.08	47°—12′	3.40	73°—37′	14.00	85°—55′
0.35	19°—17′	1.10	47°—44′	3.50	74°— 3′	15.00	86°—11′
0.36	19°—48′	1.12	48°—14′	3.60	74°—29′	16.00	86°—25′
0.37	20°—18′	1.14	48°—45′	3.70	74°—53′	17.00	86°—38′
0.38	20°—48′	1.16	49°—14′	3.80	75°—15′	18.00	86°—49′
0.39	21°—18′	1.18	49°—43′	3.90	75°—37′	19.00	86°—59′
0.40	21°—48′	1.20	50°—12′	4.00	75°—58′	20.00	87°— 8′
0.41	22°—18′	1.22	50°—40′	4.10	76°—18′	25.00	87°—43′
0.42	22°—47′	1.24	51°— 7′	4.20	76°—36′	50.00	88°—51′
0.43	23°—16′	1.26	51°—34′	4.30	76°—54′	75.00	89°—14′
0.44	23°—45′	1.28	52°— 0′	4.40	77°—12′	100.00	89°—26′
0.45	24°—14′	1.30	52°—26′	4.50	77°—28′	Inf.	90°— 0′
0.46	24°—42′	1.32	52°—51′	4.60	77°—44′		
0.47	25°—10′	1.34	53°—16′	4.70	77°—59′		
0.48	25°—38′	1.36	53°—40′	4.80	78°—14′		
0.49	26°— 6′	1.38	54°— 4′	4.90	78°—28′		

Rating of Condensers

The heat capacity of a condenser—the amount of heat that can be transferred from the refrigerant to the air—must always be greater than the tonnage rating of the compressor to which it is to be connected. Although a compressor can remove 12,000 Btu per hr per ton, it rejects to the condenser this amount, *plus* the heat of compression. For example, referring to Table 3, a "Freon" compressor operating between a suction temperature of 40 F and a condensing temperature of 102 F will reject, theoretically, slightly more than 228 Btu per minute to the condenser per ton of refrigerating capacity, or 13,680 Btu per hr.

[1]See HEATING, PIPING AND AIR CONDITIONING, January, 1938, p. 4.

Table 3—Heat Theoretically Rejected to Condenser ("Freon" Only) in Btu per Minute per Ton of Refrigeration

Evaporator Temp.	Temperature of Condensing Vapor, F										
	80	85	90	95	100	105	110	115	120	125	130
−20	258	262	266	270	275	280	284	290	295	301	307
−15	254	257	261	266	270	274	279	284	289	295	301
−10	250	253	257	261	265	269	274	279	284	289	294
− 5	246	249	253	257	261	265	269	274	278	283	289
0	242	246	249	253	257	260	265	269	273	278	283
5	239	242	245	249	253	256	260	264	269	273	278
10	235	238	242	245	249	252	256	260	264	268	273
15	232	235	238	242	245	248	252	256	260	264	268
20	229	232	235	238	241	245	248	252	255	259	264
25	226	229	232	235	238	241	244	248	251	255	259
30	223	226	229	232	235	238	241	244	248	251	255
35	220	223	226	229	231	234	238	241	244	247	251
40	218	220	223	226	228	231	234	237	240	244	247
45	215	218	220	223	225	228	231	234	237	240	244
50	213	215	218	220	223	225	228	231	234	237	240
55	210	213	215	218	220	223	225	228	231	234	237

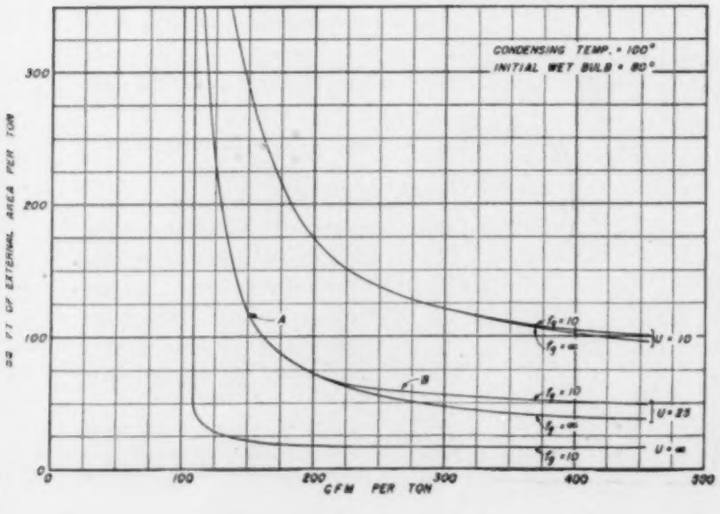
Evaporative condensers are seldom rated in terms of their actual heat capacity; they are customarily rated in terms of the tonnage of the compressor to which they may be connected. In order to find the tonnage rating of an evaporative condenser—that is, the tonnage rating of the compressor to which it may be connected—the heat capacity of the condenser is divided by 12,000 plus the heat of compression. Thus, if 13,680 Btu per hr per ton is rejected to the condenser of example 1, it may be connected to a compressor having a rating of

$$\frac{134{,}400}{13{,}680} = 9.8 \text{ tons}$$

Selection of Air Quantity and Coil Area

When a condenser is to be designed for a given load, head pressure, and initial wet bulb temperature, any desired combination of air quantity and coil area may theoretically be selected. As the quantity of air is increased, the amount of surface required is decreased as illustrated by Fig. 4. As a practical matter, however, the coil surface area and the quantity of air that should be used are confined within rather narrow limits because, as is apparent in Fig. 4, a reduction in the quantity of air below a certain limit results in an increase in surface that is out of all proportion to the decrease in air quantity. Similarly, a reduction in surface area below a certain limit results in an increase in air quantity that is out of all proportion to the decrease in surface. For the conditions of Fig. 4, if the condenser constants were $f_s = 10$ and $U = 25$, the air quantity and the surface area should be selected on the section of the curve lying between approximately points A and B. A large increase in air quantity beyond point B results in only a negligible reduction in the area of surface required.

Fig. 4—When a condenser is to be designed for a given load, head pressure, and initial wet bulb temperature, any desired combination of air quantity and coil area may theoretically be selected. As the quantity of air is increased, the amount of surface required is decreased



Curves like those of Fig. 4 are of value in studying condenser designs because they clearly show the relation between the required air quantity and coil surface. The method of computing the points on these curves is illustrated in the following example. First some value of G/A (lb of air per hr per sq ft of outside surface) is chosen at random. Then G/H, the equivalent abscissa of Fig. 4, is computed from equation 1. Multiplying A/G by G/H yields A/H, the ordinate.

Example 2: Select at random a value of $(G/A) = 20$, and compute the required area and cfm per ton. The condensing temperature of the refrigerant is to be 100 F when the initial wet bulb temperature of the air is 80 F. Assume that $f_s = 10$, and $U = 25$.

Solution:

$$M = \frac{f_s}{0.24}\frac{A}{G} \quad \text{.....[2]}$$

$$= \frac{10}{0.24}\frac{1}{20}$$

$$= 2.08$$

Interpolating in Table 1, for $M = 2.08$, $Z = 0.8749$.

$$N = \frac{GZ}{AU} \quad \text{.....[3]}$$

$$= 20\,\frac{0.8749}{25}$$

$$= 0.70$$

Referring to Table 2, for $N = 0.70$, $\theta = 35°\text{-}0'$.

Using the chart of Fig. 2, for $t_a = 100$ and $t_1' = 80$ F, $t_w = 90.7$ F.

From the psychrometric tables,[1] for $t_1' = 80$ F, $h_1 = 43.60$ Btu; and for $t_w = 90.7$ F, $h_w = 56.80$ Btu.

From equation 1,

$$\frac{G}{H} = \frac{1}{(h_w - h_1)\,Z}$$

$$= \frac{1}{(56.80 - 43.60)\,0.8749}$$

$$= 0.0866 \text{ lb per Btu}$$

$$\frac{A}{H} = \frac{A}{G} \times \frac{G}{H}$$

$$= \frac{1}{20} \times 0.0866$$

$$= 0.004330 \text{ sq ft per Btu per hr}$$

If the heat rejected to the condenser amounts to, say, 13,680 Btu per hr, the volume of air required per ton is $0.0866 \times 13{,}680 = 1185$ lb per hr per ton.

$$\frac{1185}{60 \times 0.075} = 263 \text{ cfm per ton}$$

Also, the external condensing surface required per ton is $0.004330 \times 13{,}680 = 59.2$ sq ft per ton.

[To be continued]

How to Avoid Waste in Plant Heating

By **Mortimer Freund***

Galloway

INDUSTRIAL plants require heat (1) to produce either by direct or indirect application the required physical and chemical changes in the materials processed; (2) to produce motive power; (3) to condition the atmosphere inside of the buildings so that the work can be performed under the conditions best suited to maintenance of the comfort and health of the workers and their capacity to produce efficiently and also, in special cases, so that the condition of the surrounding atmosphere is such as is required to produce the physical and chemical changes in the materials being worked upon; and (4) to raise the temperature of water required for cleansing purposes. It is common, even if not quite technically correct, to refer to the first as *process heating,* to the second as *power,* to the third as *building heating and air conditioning,* and to the fourth as *domestic water heating.*

In industrial plants heat is produced and applied either by direct exposure to the products of combustion, or through the medium of steam, heated liquids, heated air and other gases, or through the medium of electricity. We should, perhaps, add to these the heat evolved in chemical reactions. This article, however, will be confined to consideration of heating systems in which steam is the medium of heat transfer.

Heat is thus seen to be in one way or more a fundamental requirement in every industrial plant. Maintenance of the supply of heat, of its distribution throughout the plant, and of its applications, is necessary if the plant is to continue to function. Likewise the cost of this service is an important element, not always recognizable at first glance, in the direct cost of the processes of manufacturing plant product and also in the so-called fixed or overhead costs which do not vary in simple direct proportion to plant output. In times of seasonal slack or general economic depression, high overhead charges are (or should be) a matter of serious concern to the management and to the owners of the enterprise, especially when competition results in lower income from sales, and consequently the meeting of costs of production—to say nothing of securing some profits from the enterprise—requires close examination into every possibility of reducing those costs.

Heating systems should therefore be a primary concern of both management and ownership. Quite generally the operating engineer, the mechanical superintendent, or the master mechanic, who are directly concerned with operation and maintenance of the heating systems, appreciate the importance of heating in the industrial scheme—the general superintendent frequently does not. Ownership, unless it happens to be in close personal contact with the mill, factory or plant, has too often merely a hazy understanding of the importance of the heating system, which is stirred only if a breakdown interferes with or completely stops plant production. Again, it is usually in periods of economic depression that interest in reducing costs is aroused, not in periods of prosperity, and it is during just such periods that the capital expenditure necessary to secure reduction in operating and maintenance costs cannot be made. Wise industrial management with the assistance of wise industrial ownership does not permit heating systems to deteriorate or suffer loss in efficiency, but instead keeps constant check to insure proper maintenance and economical operation.

The comment may be made at this point that all that has been said so far is "old stuff," and it is. It has been the subject of talks at meetings of engineering societies and manufacturers' associations. It has been embodied in articles appearing in technical and trade publications and in books dealing with management of industries and design of industrial plants. It is spread before the public in countless advertisements by manufacturers of equipment. But it loses no force through repetition and it is doctrine that has not yet reached or been understood by all of those who stand to benefit thereby, as existing conditions in many of the industrial plants will bear witness. It should be repeated as frequently and as widely as possible.

What Troubles Are Found in Heating Plants?

What troubles are found in heating systems? What makes them inefficient and unnecessarily expensive to operate and to maintain? How can these defects be corrected in existing heating systems and how can they be avoided in new ones which are being planned? As already stated, this discussion will be confined to heating systems in which steam is the medium of heat transfer, although some of the general considerations apply equally to other types.

The ills of heating systems may be compared to those of the human body. Some are "congenital," due, however, in the case of heating systems to factors for the most part within our control—principally lack of thought and foresight in the original planning and installation. Others are due to age or deterioration, usage or wear and tear, whatever term one may wish to use. Others are due to "excrescences"—additions made without thought to

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consequences which react on the systems to interfere with their proper operation. Still others are caused by faulty operating methods.

Many of these ills can be cured, some at small cost; others (particularly some of those due to faulty planning) may require major operations to get rid of the troubles, while some are practically irremediable and would involve complete reconstruction. It is therefore essential to bear in mind such mistakes and troubles when building a new industrial plant or when purchasing or leasing buildings in which to establish one, so that consequent financial losses can be avoided.

In the class of "congenital ills" afflicting heating systems are boilers with inadequate combustion space, poor draft, not properly sized for efficient operation with varying loads, inadequate firing space and fuel storage, lack of provisions for economical fuel handling and removal of ashes, lack of headroom which compelled the installation of an uneconomical type of boiler or one difficult to maintain and keep clean, constricted boiler room space with no possibility for boiler plant expansion if needed, and lack of boiler room ventilation. Most of the above are difficult, if not practically impossible, to remedy. Forced draft blowers can be utilized to improve combustion and permit use of cheaper fuel. Some means of providing adequate ventilation can be provided.

Also in this class of troubles are those with gravity systems in which difficulty and noise occur in returning the condensate because of improperly graded piping, or because of drip points not sufficiently above the water line of the boilers; also the presence of defective air valves on radiators and pipe coils and at other points in the distributing system, which were installed because they were cheap, but whose inability to free the system of air has cost far more in excessive fuel burned than the saving in first cost. Pipes can frequently be regraded, poor air valves can be easily replaced with good ones. It is usually not feasible to alter the relation of steam mains to water line and it may be necessary to resort to pumps or lifting traps. Frequently, steam and hot water pipes and smoke connections are covered with a poor grade of covering, or even with none at all. Heat which is allowed to pass out of the system into spaces where it serves no useful purpose represents fuel wasted. Good non-conducting covering pays for itself in a short time.

In the second class of heating system troubles noted above are boiler scale, dirt and sediment which collect in traps, especially in those on the return connections of vacuum heating systems. Boilers should be inspected periodically to determine the presence of scale or corrosion, and feedwater treatment should be resorted to if necessary. All traps should be inspected at regular intervals and, after cleaning of both body and valve seat, each should be tested for tightness and reconditioned so as to close tightly in contact with steam and open promptly in contact with water. Wear in vacuum pumps is also a cause of heating system troubles and pumps should be overhauled periodically to keep them efficient and especially to remove any possibility of breakdown.

In this class also is oil which enters the heating system through faulty separators or eliminators when exhaust steam is used. Oil films which coat the inner surfaces of heating apparatus and piping materially diminish the effectiveness of these in transferring heat and also clog valves, traps, etc. A good type of oil separator should be selected in the first place and all separators should be examined from time to time, reconditioned or replaced if necessary and the heating system thoroughly cleansed.

Courtesy Trane Co.

Three unit heaters at a mounting height of 33 ft heat a new addition to the Oil Gear Co. plant, Milwaukee

In the third class of heating system troubles are drip connections from high pressure lines which introduce vapor into vacuum returns, and "temporary" or "emergency" drip connections which discharge condensate to waste and are allowed to remain so connected permanently. Also there is the use of live steam connections to help out circulation, while exhaust steam is being wasted into the atmosphere.

"Too Many Boiler Plants Spoil the Cost Records"

In this class of "excrescences" can also be included a trouble of far more costly consequences—the operation of separate boiler plants and heating systems in the same industrial plant or group, where a single plant could be made to serve. Many will recall that up to 30 years ago and possibly even later, each of the United States navy yards, or at any rate the larger ones, actually operated as many as three separate boiler plants. The larger yards are in effect industrial plants owned and operated by the government, but this condition had arisen because three distinct bureaus or departments—yards and docks, steam engineering and construction, and repair—produced and distributed the steam and electricity required for their respective functions. By consolidation of boiler and electric plants into one central plant, large reductions in operating costs were effected.

Still more striking is the case of a large industrial plant consisting of a group of 13 buildings. This establishment was served by a boiler plant consisting of seven boilers which supplied the electric generating plant, process steam for a bleachery, dye house and drying equipment.

and steam for building heating. A new activity was undertaken subsequently at one end of the establishment which also involved bleaching, drying and finishing processes. Although the boiler plant was centrally located in the group of buildings, a complete new boiler house with chimney and three boilers was built near the new department. In the meantime a heavy surplus of exhaust steam from engines and pumps was wasted to the atmosphere.

This dual boiler plant operation had continued for almost 10 years when the writer's firm was called in as consultant. By collecting all exhaust steam, extending and enlarging the exhaust steam main, utilizing exhaust steam for water heating and cutting down sudden demands for steam by the use of hot water storage heaters, converting the heating system into a vacuum return system and returning wasted drips to the boilers, it was possible to shut down the second boiler plant and operate with the original seven boilers. The second boiler house was dismantled and used for other purposes. One boiler was transferred to the old boiler house as a measure of precaution, but, as a matter of fact, it was never used, although the plant output increased steadily. This boiler plant continued to function for many years, until, because of obsolescence, the boilers were replaced by a smaller number of large units and the boiler house reconstructed and modernized.

It should be noted, however, that under special circumstances which would require an expensive steam distributing system, more than one independent plant heating system might well be justified. The point to be stressed is that in many cases multiple plants are allowed to continue to operate wastefully, or are arbitrarily installed without careful consideration of all the possibilities.

Speaking generally, industrial plants require heat for process, power, building heating and air conditioning, and domestic water heating. Thus, heat in one way or another is a fundamental requirement in every plant . . . The author maintains a broad outlook of the problem of heat for the plant, and his discussion covers its use for the four services mentioned above. He reviews errors in design and troubles in operation that lead to uneconomical plant heating, and outlines means for correcting them

The fourth class of heating system troubles involves faulty methods of operation, such as maintaining vacuum in excess of that necessary for proper results. This is likely to produce troubles with traps and valves. Use of live steam when exhaust steam is available has already been cited. Leaks at valves due to failure to repack when necessary is another. Check valves which stick should have attention and a Hartford loop seal on each boiler will eliminate troubles due to faulty check valves in the gravity return connections.

Seven Ways to Heating Economy

To conclude, in planning heating systems for industrial plants consideration should be given to the following:

1. Space and structural conditions as they affect boiler plant, supply and return piping, location of heating equipment. The building or buildings should be designed to accommodate the required heating system. The heating system should not be cramped or its operation impaired by being made to fit into the building. The requirements of the heating system should be established in the preliminary stages of planning.

2. The design should be adequate. The system should be flexible in all of its parts, capable of operating economically under varying loads and in periods of slack as well as full plant production. It should be capable of being expanded to accommodate the increased output of product which is within the reasonable possibility of factory expansion.

3. The design should be as simple as is consistent with securing economy in operation. It should be suited to the size and character of the particular industry and to the personnel which may be expected to operate it. Multiplicity of indicating, recording and integrating instruments and gages and other refinements which are appropriate and useful in a large automobile manufacturing plant, for example, may be out of place and an extravagance in a smaller plant, where they may soon fall into what the late Grover Cleveland termed "innocuous desuetude" because of lack of personnel with the time and ability to make full use of them. The extent to which refinements in control are planned should be governed by similar considerations.

4. The uses to which different sections of the building or buildings are to be put should be ascertained. Where processes demand higher temperatures than those for normal comfort, furred outside walls and insulated partitions should be considered to reduce loss of heat to the surrounding cooler spaces and waste in fuel consumption.

5. The uses of different sections of the factory will also have important bearing on the type of heat emitting surface selected. There is no sense in using radiators or pipe coils, whose effectiveness as heaters is due to a large extent on their ability to radiate, on walls in front of which bins, shelving and large machines or fixtures are to be placed. The time allowable for bringing working spaces up to the proper temperature is another factor in this selection. Departments where the operations are sedentary and especially where women operators are employed, must be brought to comfort temperature by the time work begins. Otherwise, cold hands and chilled bodies will result in curtailed production. In most plants the heating system is shut down entirely at night, or night heating is confined to those sections of the plant in which the water and plumbing systems and automatic sprinkler system have to be protected against freezing.

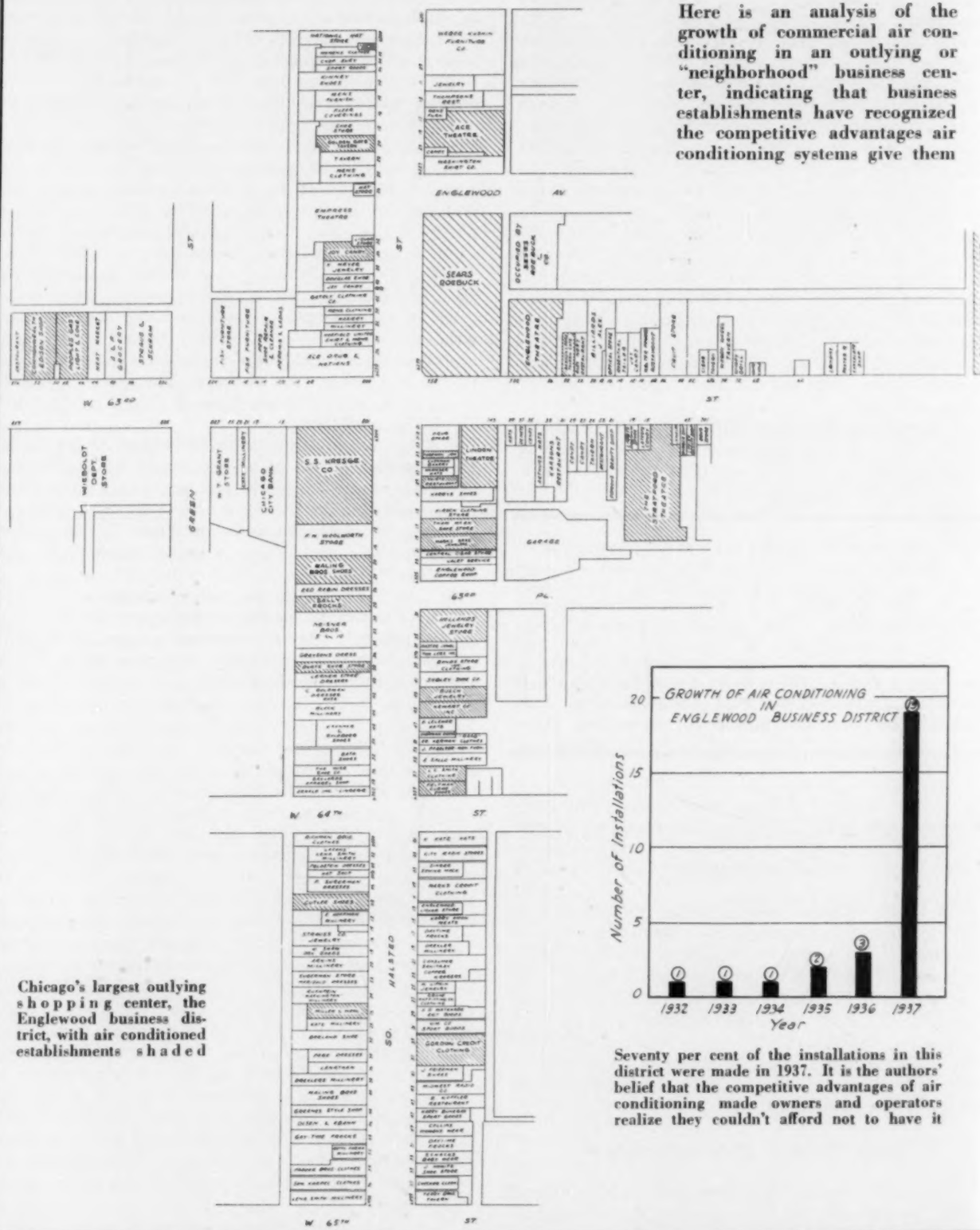
Modern unit heaters possess the advantage of concentration of large amounts of heating surface in smaller space than that required for radiators or pipe coils and the further advantage of rapid air movement and distribution of heating effect. They afford a certain degree of temperature control because the fan can be automatically shut down, or if kept in operation some of the air may be bypassed, when overheating occurs, the effectiveness as a heater being thus greatly reduced as well as the consumption of steam. They are also easily relocated. On the other hand, electricity is consumed as long as the fan is in operation.

[Concluded on page 173]

Air Conditioning Becomes Locally Competitive

By Knight C. Porter* and William P. Rock*
*Air Conditioning Division, Commonwealth Edison Co.

Here is an analysis of the growth of commercial air conditioning in an outlying or "neighborhood" business center, indicating that business establishments have recognized the competitive advantages air conditioning systems give them



Chicago's largest outlying shopping center, the Englewood business district, with air conditioned establishments shaded

Seventy per cent of the installations in this district were made in 1937. It is the authors' belief that the competitive advantages of air conditioning made owners and operators realize they couldn't afford not to have it

THE corner of 63rd and Halsted streets is probably the busiest and largest of Chicago's outlying or "neighborhood" shopping centers. Focal center of the Englewood district, it is situated six miles south and one mile west of the "downtown area." It is said that more than 1,000,000 people enter the stores, restaurants, and theaters of this district each week.

The people of Englewood and vicinity had their first local experience with air conditioned comfort in the summer of 1932. This they found in the new Southtown theater which opened December 25, 1931. In 1933, the district boasted a second installation, this time in a restaurant. In 1934 the third was installed in a department store. The year 1935 saw the addition of two more in a five-and-ten cent store and in a shoe store. In 1936, air conditioning began to make sizeable inroads with two more theaters and a candy store.

During the first eleven months of 1937 there were 19 new installations in the few blocks of this compact business district. This means that 70 per cent of the existing 27 installations were sold in 1937. Many types of business are represented in this group of installations, as follows: clothing stores, 5; shoe stores, 5; theaters, 5; jewelry stores, 4; restaurants, 3; department stores, 2; appliance stores, 2; and candy stores, 1.

From the foregoing picture of the air conditioning growth of an outlying commercial center, what deductions and predictions may be made?

First, since the 1937 increase in installations in this shopping center is of the nature of 200 per cent as compared with but about 35 per cent for the city as a whole, we recognize that certain unusual local sales stimulus accounts for this large increase. It is our belief that the competitive advantages of air conditioning in Englewood had arrived at such a point in 1937 that *many commercial operators suddenly recognized that they couldn't afford not to have air conditioning.* We feel that the chart of growth tends to point to the conclusion that the momentum of local competition (insofar as customer comfort is concerned) expedited acceptance of air conditioning.

Although we are now talking about a total of 27 installations, commercial saturation is still quite low—there remain about 150 business places in the vicinity which are not yet air conditioned. It is our belief that 1938 air conditioning sales in this section will greatly overshadow those of 1937. Why? Simply because the competitive necessity of those merchants who are not yet prepared to offer summer comfort is more acute than ever.

A sectional study such as this may well be undertaken for a number of local shopping centers in the near future. It is thought that the preparation of this sort of study should be of interest to the merchants in the various shopping centers. It should allow them to crystallize their thoughts in relation to air conditioning. The results of such a study may also suggest a new manner of air conditioning sales approach and promotion. In any event, we feel that probably the greatest air conditioning sales stimulus in 1938 will be local commercial competition.

Plant Heating—

[Concluded from page 171]

6. The provision of some form of temperature control. This is true both of existing heating systems and those which are being planned. Over-heating as well as under-heating interferes with plant production by rendering those at work uncomfortable, distracting their attention from their work, impairing their ability to produce and hence resulting in poorer quality as well as quantity of product. In addition, there is a direct waste of heat.

Many Developments in Controls

Probably the most striking developments in heating systems in recent years have been in the field of temperature control. This has been achieved by several methods, including (1) varying the steam pressure or the vacuum, or both, where vacuum heating systems are used; (2) using orifice plates in the inlets to radiators and coils sized to admit just enough steam to fill the radiator completely in cold weather under the maximum pressure used and only enough to heat the radiator partially during mild weather under reduced steam pressure; and (3) interrupting the supply of steam to the system at intervals, the length of the periods of admission and shut-off being varied either manually or automatically as desired. All of these methods follow the principle of controlling the effectiveness of the heating surface as a whole.

Another method used is that of zoning, in which the heating surface is divided into groups, selectively controlled from one point, so that certain departments only can be heated at times when other departments do not require heat and also so that during milder weather less steam or none at all can be supplied to the less exposed sections of the buildings, while only those sections exposed to the prevailing cold winds are kept fully heated.

It is always worth while to investigate the possibilities of introducing some form of temperature control into the heating systems of existing industrial plants.

7. Provision should be made for water heating and storage of hot water, especially for use in industrial processes. The practice of blowing live steam directly into cold water should be minimized or avoided altogether, as this leads to sharp boiler load peaks, which result in uneconomical operation and over-sized boiler plants. This is particularly true in laundries, dye houses and bleacheries. Every attempt should be made by use of exchangers to reclaim heat from process liquids which are being discharged to waste. Likewise heat from wasted gases can be reclaimed to preheat water used in industrial processes. This can frequently be accomplished after investigation in existing plants and should certainly be considered in planning new ones.

Air conditioning and exhaust ventilation, which are intimately tied up with the heating system, play a constantly growing rôle in industrial plants, not only in providing comfortable conditions for the workers and hence in improving the quality and the quantity of the product, but also as necessary attendant factors in certain industrial processes. The problems encountered both in design and operation of such systems constitute a separate subject for treatment in detail.

How to Use

Reynolds' Numbers in Piping Design

By G. A. Gaffert*

The newer approach to piping design through the use of Reynolds' numbers involving viscosity of steam and water is timely. With the advent of higher pressures and particularly temperatures, the older formulations are subject to greater errors, whereas research data on viscosity of steam, water, and various gases provide a more exact analysis of frictional flow. The mechanical engineer can well make use of this contribution to the flow of fluids

I WAS most impressed with the approach to the problem of friction of fluids in pipes when I took a course in chemical engineering unit operations at the University of Michigan. The fundamental approach of the chemical engineer necessarily is different from that of the average mechanical engineer, for the former has to deal with a wide variety of fluids, solutions, and gases, which must be transported in various industrial processes. Usually the head which must be developed by the transfer pumps is only a matter of static head and friction, and the latter is no insignificant part of the total.

The chemical engineer found at an early date that he could not rely upon the work done by mechanical engineers in establishing empirical formulas for the flow of water, air and steam. Add to the above the temperatures of fluids and vapors encountered in process work and you have a picture of the chemical engineer's situation.

Let us survey the research work available relative to the flow of fluids and gases. For the flow of water in pipes we have the work of Chezy, D'Arcy, Hazen and Williams, and Fanning—developed in the most part for hydroelectric practice. For the flow of air we have the various researches of laboratories at the large universities and the test laboratories of several fan manufacturers. In the case of steam I would refer you to the work of Unwin, Carpenter, Babcock and Fritzsche. All of these formulations contain experimental constants to make the equations agree with actual observations. Most of these equations take no cognizance of the variation of fluid behavior due to temperature.

We may digress at this point and discuss viscosity. Viscosity can be defined as that property of a fluid which enables one layer of it to resist the sliding of an adjacent layer over it. This may be portrayed as shown in Fig. 1. Imagine a cube of jelly with a horizontal

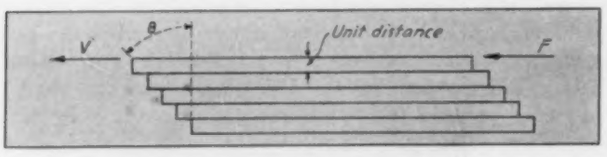


Fig. 1—Sketch portraying concept of viscosity

force F applied at one edge. As the magnitude of the force increases, the cube deflects and layers begin to slide, each relative to its adjacent layer. Finally, as Newton observed, dynamic equilibrium is reached between the force and the velocity gradient.

Newton called this ratio μ. The proportionality may be written,

$$\mu = \frac{\text{Shearing force}}{\text{Velocity gradient}}$$

In c.g.s. units,

$$\mu = \frac{\text{dynes} \times \text{seconds}}{\text{centimeters}^2} = 1 \text{ poise (by definition)}$$

Unfortunately for the engineer, much of the research work done abroad and even in this country is in the c.g.s. (centimeter-gram-second) system. It is therefore necessary to convert viscosity data in centipoises (1/100 of a poise) to English units. In English units viscosity is expressed as lb/(ft)(sec). Since 1 gram = 1/454 lb and 1 cm = 1/30.48 foot,

$$1 \text{ centipoise} = (1/100) \times \frac{\frac{1}{454}}{\frac{1}{30.48}} = 0.000672 \text{ lb/ (ft) (sec)}$$

Osborne Reynolds made several observations on the flow of fluids and we are indebted to him for the establishment of the critical velocity in flow of fluids. He pointed out that below a certain velocity, *laminar* or *viscous* flow occurred, and above this critical velocity *turbulent* flow obtained. The boundary between viscous and turbulent flow may be determined by the following expression, which is called Reynolds' criterion:

$$\frac{DV\rho}{Z} = 2000 \text{ to } 2200$$

where D = inside pipe diameter, feet,
V = velocity, feet per second,
ρ = fluid density, pounds per cubic foot,
Z = viscosity, lb/ (ft) (sec).

It will be noted that the above criterion is dimensionless.

A few typical applications will demonstrate that practically all modern applications lie in the turbulent region. Fig. 2 shows a plot of $DV\rho/Z$ versus unit friction or $\phi(DV\rho/Z)$. The curves shown here have been determined from plotting thousands of experimental points, the

*Mechanical Engineer, Sargent and Lundy, Inc. (From a recent talk before the Chicago section of the American Society of Mechanical Engineers).

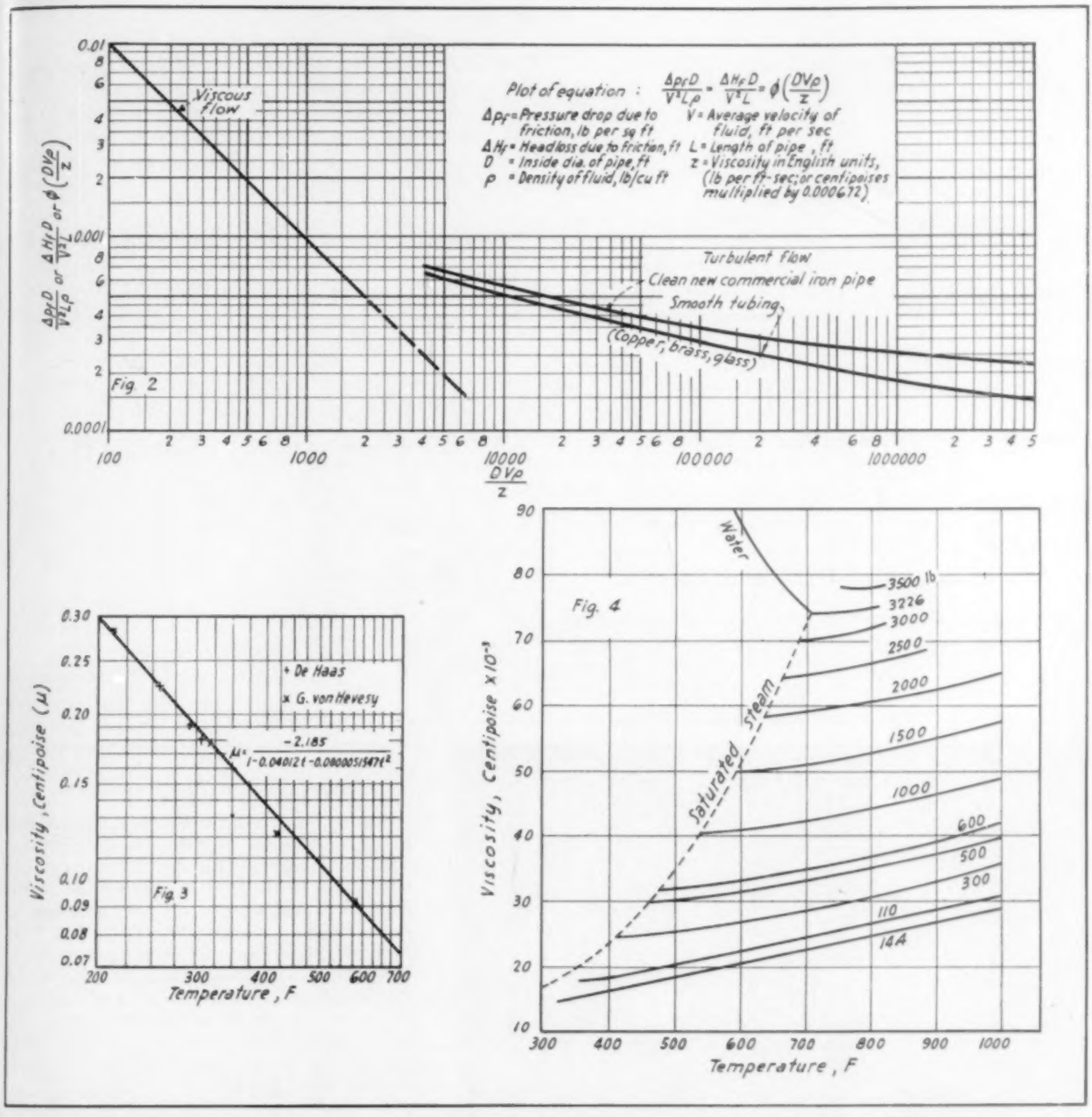


Fig. 2—Plot of Reynolds' criterion. Fig. 3—Viscosity of water (from International Critical Tables). Fig. 4—Viscosity of steam (from work of Purdue University, Speyerer, and Schiller)

only distinction being that for commercial iron pipe and smooth tubing. These curves may be considered as well established to a value of $(DV\rho/Z) = 5{,}000{,}000$. The discontinuity between viscous and turbulent flow is evident.

To apply Reynolds' criterion in the determination of friction it is necessary to have reliable data on the viscosity of fluids. Fortunately, fluid viscosity is not greatly affected by pressure, and in most applications a curve of fluid viscosity versus temperature only is necessary. Fig. 3 shows the variation of viscosity of water with temperature as reproduced from the International Critical Tables. Water at a temperature of 69 F has a viscosity of 1 centipoise. The viscosity of air and several permanent gases may also be taken from the above mentioned tables.

Experimental work at Purdue University has brought up to date the existing data by Speyerer and Schiller on the viscosity of steam, the data now being complete up to 2000 lb absolute and 1000 F. With some extrapolation the data may be used up to the critical pressure and 900 F. It is to be noted from Fig. 4 that pressure has a considerably greater effect on the viscosity of steam than does temperature.

Solving a Problem in Water Flow

Let us use these data in a problem involving water flow. Assume that a boiler feed pump removes feed water at a temperature of 200 F from a deaerating heater and pumps it at a rate of 100,000 lb per hr to a boiler.

The length of the line, including the equivalent resistance of valves and fittings, is 400 ft, and the pipe is 4 in., standard weight, with an inside diameter of 4.026 in. Density of water at 200 F is (from steam tables) 60.12 lb per cu ft.

$$\text{Volume flowing} = \frac{100{,}000}{3600} \times \frac{1}{60.12} = 0.462 \text{ cu ft per sec}$$

$$\text{Velocity} = \frac{0.462 \times 4 \times 144}{(4.026)^2 \times \pi} = 5.23 \text{ ft per sec}$$

Reynolds' number (from Fig. 3) for water at 200 F = 0.3 centipoises = 0.3×0.000672 in English units

$$\text{Reynolds' criterion} = \frac{DV\rho}{Z} = \frac{(4.026)\ (5.23)\ (60.12)}{12 \times 0.3 \times 0.000672} = 522{,}000$$

$$\text{Value of } \emptyset \frac{(DV\rho)}{Z} = 0.000275 \text{ (from Fig. 2)} = f$$

$$\text{Friction drop, } \Delta h_f = f\left(\frac{V^2 L}{D}\right) = \frac{0.000275 \times (5.23)^2 \times 400 \times 12}{4.026}$$

Friction drop = 8.97 feet of water

Analysis Applied to Steam Flow

Reynolds' analysis may also be applied in the case of a problem in steam flow. Let us check the pressure drop in a steam lead from a boiler with 135 ft of equivalent pipe, using the following conditions: Steam pressure, 425 lb gage; steam temperature in line, 825 F steam flow, 65,000 lb per hr; 6 in. Schedule 80 pipe with an outside diameter of 6.625 in. and an inside diameter of 5.761 in. Specific volume of steam is (from steam tables) 1.59 cu ft per lb.

$$\text{Velocity} = \frac{65{,}000 \times 1.59 \times 144 \times 4}{\pi \times (5.761)^2 \times 60} = 9525 \text{ fpm} = 158.6 \text{ fps}$$

$\mu = 33 \times 10^{-3}$ centipoises (from Fig. 4)

$= 0.000672 \times 33 \times 10^{-3} = 2.2176 \times 10^{-5}$ lb/ (ft) (sec)

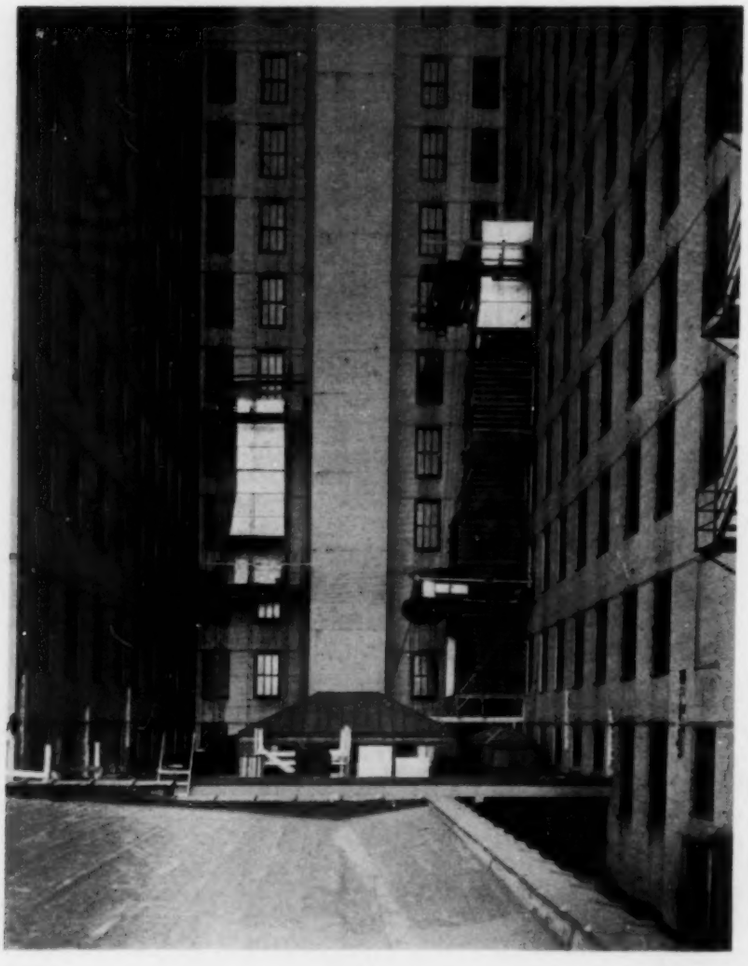
$$\text{Reynolds' number} = \frac{DV\rho}{Z} = \frac{5.761 \times 158.6 \times 10^5}{12 \times 1.59 \times 2.2176} = 2{,}160{,}000$$

$$\text{For } \frac{DV\rho}{Z} = 2{,}160{,}000,\ \emptyset\left(\frac{DV\rho}{Z}\right) = 0.00024 \text{ (see Fig. 2)}$$

$$\frac{\Delta p_f D}{V^2 L \rho} = 0.00024$$

$$\Delta p_f = \frac{(0.00024)\ (158.6)^2\ (135)\ (12)}{(5.761)\ (1.59)} \times \frac{1}{144}$$

Pressure drop = 7.42 lb per sq in. for the length of 135 ft



Union Pacific Air Conditions

Two air ducts under construction at the Union Pacific's headquarters building, Omaha, Nebraska, which will convey air for the 12 floors from and to a central dehumidifier in a pent house in the court. The design provides for one zone unit in conjunction with five zone dampers for each floor. Each of the 12 floors will have one booster fan with control to regulate static pressures in the five zone ducts. Total air circulation is approximately 200,000 cfm, of which 90,000 cfm will be outside air. Volume of space to be cooled is 3,600,000 cu ft and the floor area is 300,000 sq ft. Ultimate occupancy is 3000.

Twelve radial type compressor units will provide 1800 gpm of chilled water to the central dehumidifier. Eleven of the 12 condensing units will be duplex machines consisting of two compressors of seven cylinders each direct connected by flexible couplings to a single 75 hp, 1750 rpm, 208 volt motor. The other is a 100 hp duplex seven cylinder machine connected to a 100 hp, 1750 rpm motor. The complete plant, with a 900 ton rating, will require a total of 1461 hp, of which 150 hp is standby pumps.

Refrigerant and water piping is designed so that any number of machines may be operated, with evaporators interconnected on the water side and independent on the refrigerant side to allow water to circulate from a common header to all coolers and the refrigerant to be confined to individual evaporators. Condensing water is to be cooled by a forced draft spray type cooling tower on the roof designed to cool 2500 gpm from 97 to 87 F with an outdoor wet bulb of 80 F.

Contractors are the Sidles Co.

22% Less Steam Heats Office Building

By William S. Downs*

It's an old gag that if all the percentage savings the operating engineer is advised to make in the heating of his building were possible, he could end up with more fuel or steam than he started with. However, good equipment maintained and operated in an efficient manner by competent engineers can and does result in worthwhile savings in heating cost—which is the point Mr. Downs makes here. . . . His article is a running story of how the steam consumption rate for one building has been reduced, and in it he tells some of the difficulties met and how they were solved

DURING the 1933-34 heating season our steam consumption rate was 0.7030 lb per 1000 cu ft of building per degree day; last season it was 0.5467 lb. The difference represents increased efficiency in operating the system and is accountable to a change in the method of supplying steam to the system, and elimination of leaks in pipe, valve, trap and expansion joint connections.

Previous to altering our method of steam control we had been operating at anywhere from 1 to 5 lb gage pressure. It was difficult to determine the most efficient pressure range due to the difference in Btu demand of the direct radiation and indirect radiation systems. Both systems were individually controlled by separate feed lines and double stage reducing valves, but both were tied into a common return some distance from the vacuum pumps.

The Ventilating and Heating Equipment

It was at first thought that the pressure on the indirect system should be carried at a higher point than on the direct system due to the greater heat exchange rate of the blast coils. Our ventilating system is the straight intake type with no provision for recirculation of the air. This naturally entailed a high rate of Btu exchange, variable according to the severity of the weather. Whenever the outside temperature dropped to zero this rate was 1.459 Btu per cu ft of air heated to 70 F.

Our direct system is rather complicated in that up and downfeed risers are interconnected to the return lines which extend from the basement to the roof. There are 22 upfeed risers leading from the main steam header and ending at the 30th floor setback. Nine of these continue on to the 36th floor setback which is at the base of the tower. The tower radiators are fed by a line running up to the 46th floor ceiling and branching into a one way loop around the tower. Each downfeed riser is equipped with a hand valve below the loop but the main tower valve is controlled from a panel in the engineers' office in the basement.

The 22 upfeed risers and main tower line valves are pneumatically operated by three way lever cocks mounted on the control panel. It is possible by this arrangement to isolate vertical sections of the building for partial heating after regular office hours. The downfeed risers, however, cannot be isolated without crawling through an access door into the ceiling. For this reason, a good deal of lost time was entailed whenever a single circuit was to be tested for leaks. It is impossible to test a single supply and return circuit without including sections of another, or in some cases, two or three circuits.

While testing we also ran into difficulties with the indirect piping layout. It was necessary to isolate each blower circuit from the main supply line because the main return is not equipped with a cutout valve. Each blower coil is equipped with a hand return valve and by shutting off all return valves with the exception of the circuit under test we could fill the entire supply line with compressed air.

The main headers for both direct and indirect systems are 14 in. in diameter and branch immediately into smaller lines to various sections of the building. In the case of the indirect system, the 14 in. header branches into a 10 in. line running up through the building and a 5 in. line that feeds several blowers on the lower floors. The 10 in. line is reduced at the 21st floor to 8 in. and again at the 46th floor to 6 in. There are no blowers between the 25th and 48th floors and it is practically never necessary to operate any blowers above the 4th floor after regular working hours. Consequently, when the upper blowers were shut down, the whole line above the 4th floor was in a sense a huge condenser.

Of 22 blowers, 15 are equipped with blast coils, four are small recirculating blast blowers for the main floor entrance lobbies and the remainder are straight ventilating fans with no heating units. Six of the larger blowers are situated below the main floor and supply indirect heat and ventilation to basement spaces. These spaces are freely tenanted but entirely serviced by indirect heat. As a result, it is very often necessary to run the blowers below the main floor after regular office hours. There was, however, no cutout valve above the branch lines to these blowers and whenever overtime heat was required steam had to be admitted to the tempering stack line all the way to the roof.

New Valve Pays for Itself

There are 232 ft of 10 in., 296 ft of 8 in. and 76 ft of 6 in. pipe in this line above the fourth floor which is equivalent to over 1300 sq ft of external pipe surface. The line was well insulated but it seemed a wasteful practice to turn steam on the entire system when none would be used above a branch line just below the fourth floor. Accordingly, a 10 in. pneumatic valve was installed.

*Assistant Engineer, Irving Trust Co. Building, New York, N. Y.

Since making this installation it is estimated that the saving in steam amounts to approximately 67,000 lb a month. Our steam is purchased from the New York Steam Corp. and at the average rate the saving is equivalent to $60. The valve cost $210 and we paid a welder $40 to cut the 10 in. line and weld on a pair of flanges. Our own crew set the valve in place and ran the ¼ in. air line to a spare lever cock on the control panel in the basement. The entire cost of the installation including the 130 ft of air line was $275 and we saved that amount in a little more than four months.

Our vacuum equipment consists of three turbine driven venturi nozzle type pumps which could be expected to pull 25 in. at the suction inlet valve. The whole piping layout was designed with the idea of operating at pressures above zero gage but we set out to establish the lowest pressure range conformable to efficient and comfortable heating. We began by experimenting with the double stage reducing valves, adjusting them to points below zero. There was much to be said for thus cutting the pressure but some interesting as well as unique discoveries came to light.

Some time during the second year of operation the vacuum pumps began to hunt and the readings fell off to 15 in. An inspection of the impellers showed signs of corrosion and pitting. A chemist was called in to analyze the condensate and his report gave the presence of oxygen as cause for the corrosion. The impellers were replaced but the vacuum readings were very little improved.

It was not until the 1934-35 heating season that we began experimenting with the reducing valves. During the intervening season we struggled along with a condition that affected the heat supply to some sections of the building. After dropping the pressure, however, the real trouble began to be suspected. Then the vacuum fell off to 7 in. and with light loads even as low as 3 or 4 in. But we had by this time learned that the system could be operated at lower pressures and the idea that a leak had developed somewhere in the system was the ultimate conclusion.

In the meantime, it was decided that our sluggish reducing valves were a drawback to the efficient expansion of steam into the low pressure headers. These valves surged almost continuously and the tracer pencil on the flow charts surged accordingly. The recorded flow rate was a baffling maze of up and down lines that formed a black path on the chart more than 1½ in. wide and it was impossible to gage the flow of steam except by the unit integrator readings.

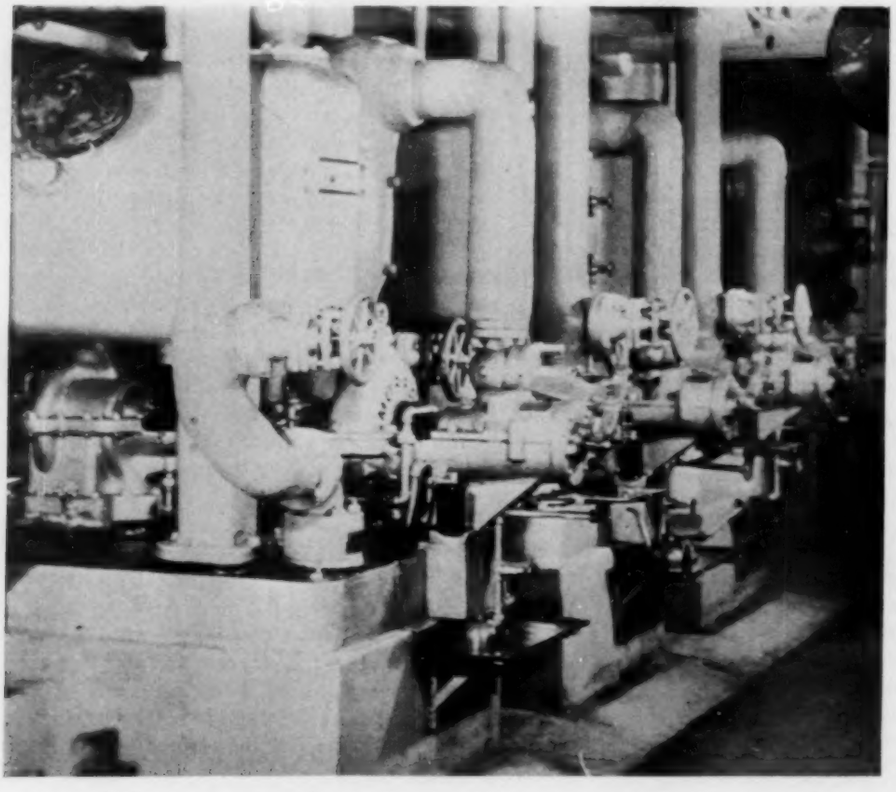
For the past two seasons we have been using a new type of heat control that in a single step reduces our high pressure steam to any desired level. This control expands the steam from 125 lb gage to our present working pressure, which is seldom above 9 lb absolute. The vacuum pumps maintain 15 in. up to any radiator in the building and we vary the differential according to the severity of the weather. This, however, is purely optional. It is very seldom necessary to spread the differential above 2½ in.

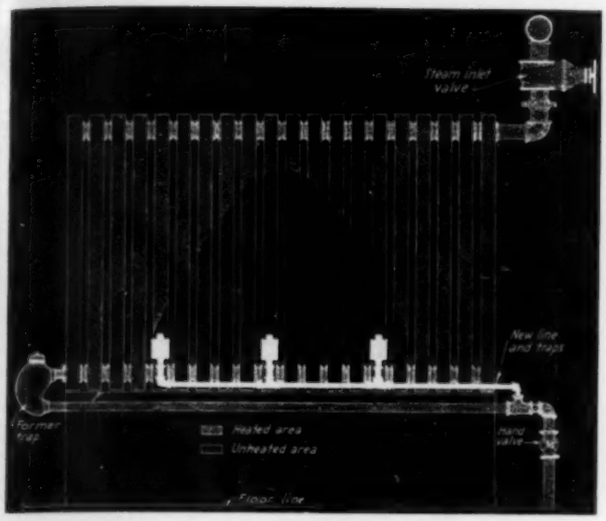
A Leak Is Suspected

After installation of the new control and the tests that followed we were certain that the system had a bad leak. It is comparatively simple to trace out a leak in the heating system of a 10 room house but the intricate network of pipe lines and expansion loops in the walls and ceilings of a 50 story skyscraper defy inspection. Night after night for a month we crawled through hanging ceilings and squirmed under duct work until the leak was located at a radiator on the 49th floor. Due to expansion of the lines, or possibly settling of the building walls, a ¾ in. line had snapped off at an elbow joint embedded in the concrete floor filling. We made the discovery at two o'clock on a cold February morning and in a couple of hours had chopped away the concrete and plugged the line.

The vacuum picked up immediately to 15 in. at the pumps which gave us 12 in. at the radiators, the difference representing a lift at the bottom of the return line which is responsible for a 3 in. drop between the pump suctions and the returns from the building. After plugging the leak we were able to operate

Our vacuum equipment consists of three turbine driven pumps





Sketch of indirect heating coil showing new method of trapping to eliminate unheated area

fairly successfully at from 10 to 12 lb absolute pressure. The system was finally and certainly down below zero gage but we were not through with discoveries. There was still some question as to the efficiency of the indirect system. In mild weather it was comparatively easy to heat the incoming air to a comfortable temperature but when the thermometer got down around 20 F it was impossible to bring the delivery air up to 70 F.

Indirect Heating Coils Partly Cold

Our blast coils are the cast iron, extended surface type. From 10 to 24 coils are connected three rows deep to separate steam and return lines and the heat is controlled by a regulating thermostat. As the air passes over the coils it should be thoroughly heated to the setting. An inspection of the heaters, however, revealed that even when the thermostat was wide open only two-thirds of the coils were heated. The rest of the heating surface was as cold as the incoming air.

The unheated area was about equivalent to an irregular 3 ft triangle with the topmost point some 12 in. from the steam channel in the center of each row of coils. The steam entering at the top is induced to flow downward through each coil to the return channel by the vacuum maintained at the outlet trap connected to the opposite end of the heaters. Blast traps guard the outlets and keep the coils free of condensate. But conceding that the trap was of sufficient capacity, nevertheless the total area of the openings was not great enough to allow proper vacuum conditions.

Actually, the 1¼ in. return line is of more than adequate size to handle the condensate and maintain peak vacuum on every internal square inch of heating surface. An analysis of the circumstances, however, seemed to indicate a choice between two alternative solutions:—The blast traps could be replaced by a full capacity bellows type trap, or we could install additional small traps.

Installing additional traps seemed at the time the best plan and we drilled into two coils in the unheated area of each row for a ⅜ in. line. A common bellows type radiator trap was installed on each of these lines and connected to the return on the pump side of the 1¼ in. blast traps. We were experimenting, and though the immediate results were gratifying, the whole affected area did not clear up. To definitely overcome the condition it was necessary to build manifolds for nine extra traps, three to each row of 20 coils. This divided the heaters into what might be called four trap sections of five coils and the grand total of 12 traps was sufficient to bring the equivalent direct radiation values up to par.

Reference to the sketch of one row of coils will convey a good idea of the condition before and after we made the changes. The 24 coil heater is our largest and for this particular installation we were obliged to add 12 new traps. The coils are four rows deep and each row was divided into five trap sections. In the 20 coil heaters, of which we have the greatest number, the coils are only three rows deep and it was only necessary to divide each row into four trap sections.

Small Leaks Affect Radiators

There is a considerable difference in using steam expanded below zero gage and saturated steam at pressures above atmosphere. This was soon determined when due to small leaks we found that the heat emission of various radiators was affected. We did not immediately learn to distinguish between the effect produced by an external leak, or the presence of water pockets in horizontal lines pitched in the wrong direction. Out of more than 1900 radiators we found possibly a dozen affected by water pockets but there were many more than this the heat emission of which was lowered by small leaks. Expansion joints, too, had to be repacked and drip traps overhauled before we could claim a fair measure of success. But at the end of the 1935-36 heating season the steam consumption rate had decreased to 0.5510 lb per 1000 cu ft of building per degree day.

Exception to this figure might be made in view of the indeterminate supply of heat from the indirect heating coils during the time we were making the trap changes. There were times when skepticism threatened the continuation of our plans to find the most efficient operating pressure range. But the gradual improvement of the general heating conditions in the building was becoming more apparent toward the end of the heating season. During the following summer months tests on risers and return lines were continued. By connecting a small line from the air compressors to the main steam header we were able to test any section of either system. A little oil of peppermint injected into the line ahead of the compressed air was a great help in detecting the tiniest leak. We found two radiators that were cracked in casting seams through the aid of this strong-smelling extract.

Job Not Done

Although the system is in very good condition now there are several additional changes that might be made further to increase our efficiency. These will come from time to time and it is hoped that our steam consumption rate may be brought to the level which we calculated it can be—less than 0.5000 lb per 1000 cu ft of building per degree day. This is not an impossible realization as long as the following rule is kept in mind:—Every decrease in the steam consumption rate is attributable to good equipment maintained and operated in an efficient manner by competent engineers.

Standardization in the Valve and Fittings Industry

By **John J. Harman***

STANDARDIZATION has become an important factor in many industries where modern mass production methods have accomplished so much in lowering the cost of desirable products to levels within the reach of large masses of the consuming public.

It has enabled manufacturers to reduce waste and save money in many ways. Reduction of unnecessary variety in manufactured product, which is accomplished by standardization, naturally results in a larger demand for the remaining types and sizes on which production is concentrated. The longer runs and fewer set-ups increase plant efficiency; fewer machines, jigs, fixtures and tools are needed; substantial savings are made in the purchase of raw materials; the cost of catalogs and sales literature is reduced; goods are packaged with less expense; and storage and warehouse costs are decreased with fewer items to handle and less money tied up in slow moving inventories. Distributors also benefit by smaller stocks and more rapid turnover; and consumers, due to the lower prices, have more money available to satisfy additional wants, thus adding to the total business volume.

Application Local at First

When standards originally began to come into use, it was natural that they should be developed first by individual companies, such standards, of course, applying only to their own products and being retained as jealously guarded secrets within their own organizations. However, as production, transportation and distribution facilities improved and commerce became more extensive and complex, efficient production and marketing required that many of these individual company standards should be unified, and this demand led to standardization by industrial trade associations, such as the Manufacturers Standardization Society of the Valve and Fittings Industry, and by technical societies, such as the American Society of Mechanical Engineers. Later, to afford representation to consumers, distributors and general interests as well as producers, thus assuring wide acceptance and use of the standards, national standardizing bodies came into existence. This latter class of standardizing bodies is represented in the United States by the American Standards Association. As time goes on, standardization on an international scale, which has already progressed to some extent, may be expected to take on added importance.

In the valve and fittings industry the growth and development of the standardization movement has been particularly notable during the period since the beginning of the present century. Naturally in connection with the rise and expansion of a fundamental development such as this, it is to be expected that there should be some differences of opinion and arguments, pro and con, on various phases of the problem. The valve and fittings industry has been no exception to this general rule, and experience has shown that in order to avoid serious complications the development of standards must be done intelligently by persons thoroughly familiar with all aspects of the situation.

Wholesale standardization involving unnecessary restrictions might very easily be carried so far as to smother the divine spark of creative genius in designers and thus put an end to progress and development. Then too, as the primary function of standardization is economy, standards should always conform as closely as possible to existing commercial practice in the industry. The total investment in patterns, tools and equipment for manufacturing existing designs of products is enormous and consideration should always be given to the economic loss that would be occasioned by unjustified mandatory unification of all detailed dimensions of a given product. Standardization proposals often involve a potential economic loss out of all proportion to the possible benefits.

In spite of differing opinions, however, numerous standards have been set up in the valve and fittings industry which are of great value to all concerned. The progress made along these lines unquestionably has been due in large measure to the continuous existence during the past thirty years of a representative organization in the valve and fittings industry devoted exclusively to the study of standardization problems.

Leaders in the valve and fittings industry realized the need of cooperative action in standardization matters at an early date, and in 1908 affected an organization known as the "Committee of Manufacturers on the Standardization of Pipe Fittings and Valves." The purpose behind this organization has always been both constructive and defensive. The trained engineers of the valve and fittings industry, who are familiar with and take into consideration all the manufacturing and economic phases of the situation, are a powerful factor in guiding standardization proposals along practical lines.

The early efforts of this committee were a very important factor in the standardization of the connecting ends of valves and fittings to insure interchangeability. It was largely through the efforts of this organization, in cooperation with standardizing bodies such as the A.S.M.E. and others, that practical standards were developed and put into use for pipe threads and flanged joints for cast iron and bronze products. In this connection it is of interest to note that although the Brigg's standard for pipe threads was projected in 1886 it was not until 1915, in consequence of some years of careful study and work by this committee, that thorough interchangeability in the joints of pipe threaded products became a broadly accomplished fact.

In 1924 the old "Committee of Manufacturers on the

*General Secretary, Manufacturers Standardization Society of the Valve and Fittings Industry, and Secretary, A.S.A. Sectional Committee B16 on Standardization of Pipe Flanges and Fittings. Member of Board of Consulting and Contributing Editors.

Standardization of Pipe Fittings and Valves" was reorganized under the name of "Manufacturers Standardization Society of the Valve and Fittings Industry" (abbreviated "M.S.S.") At that time in order to provide a definite official procedure for handling the details of the society's work, a new constitution together with a code of procedure and declarations of policy were developed. The organization as now set up is designed to provide representation of the valve and fittings industry on outside standardizing bodies in order to make sure that engineering and manufacturing problems are given due consideration in the development of standardization projects affecting the industry.

Carrying out this policy the society has become one of the active member bodies of the American Standards Association. The society also is a member of the American Society for Testing Materials and has established contacts involving workable cooperation with a number of other standardizing bodies where official representation is not feasible.

Active Standards Now in Effect

During the period from 1924 to date, the M.S.S. has developed and promulgated a number of M.S.S. "standard practices." A list of the standard practices now in effect is shown in Table 1.

From Table 1 it will be noted that seven of the standard practices shown bear a 1937 date of issue, indicating that growth in the standardization movement is very active at the present time. M.S.S. standard practices, after suitable trial in practical use for a period of time, are often developed into American Standards by the American Standards Association which, of course, affords them much wider distribution and more general recognition.

It is of interest in this connection to list the American Standards now in effect, that have been developed and promulgated by A.S.A. Sectional Committee B16 on Standardization of Pipe Flanges and Fittings (see Table 2). Sectional Committee B16 operates under the joint sponsorship of the Heating, Piping and Air Conditioning Contractors National Association, the A.S.M.E., and the M.S.S.

Table 1—List of M.S.S. Standard Practices

Number	Subject
SP-2-1937	M.S.S. 150, 250 and 300 Lb S.P. Bronze Flange and Flanged Fitting Standard
SP-3-1929	Roughing-In Dimensions for Light Radiator Valves, Union Elbows and Return-Line Vacuum Valves
SP-5-1937	M.S.S. Bypass Size Standard
SP-6-1929	Finishes for Contact Faces of Connecting End Flanges of Ferrous Valves and Fittings
SP-8-1929	Cast Iron Screwed Drainage Fittings
SP-9-1929	M.S.S. Spot Facing Standard
SP-10-1932	M.S.S. 125 Lb S.P. Bronze Screw Pipe Fittings Standard
SP-11-1930	M.S.S. 250 Lb S.P. Bronze Screw Fittings Standard
SP-16-1937	M.S.S. Pipe Bushings Standard
SP-17-1937	M.S.S. Locknut Standard
SP-20-1936	M.S.S. Specifications for Steam Bronze Castings for Valves, Flanges and Pipe Fittings
SP-21-1930	A.S.T.M. Specifications (A126-30) for Gray Iron Castings for Valves, Flanges and Pipe Fittings
SP-22-1937	M.S.S. Specifications for Malleable Iron Castings for Flanges, Pipe Fittings and Valve Parts
SP-25-1936	M.S.S. Standard Marking System for Valves, Fittings, Flanges and Unions
SP-28-1937	M.S.S. Drain Tapping Standard
SP-29-1932	M.S.S. Screw Thread Standard for High Pressure and Temperature Bolting
SP-31-1932	M.S.S. 300 Lb S.P. Malleable Iron Screw Fittings Standard
SP-32-1937	M.S.S. Ferrous Flanged Valve Center to Face Standard
SP-34T-1932	M.S.S. Service Ratings and Test Standard—Valves and Fittings
SP-35-1936	Railing Fittings—Male and Female Threads

Table 2—American Standards Developed by Sectional Committee B16.

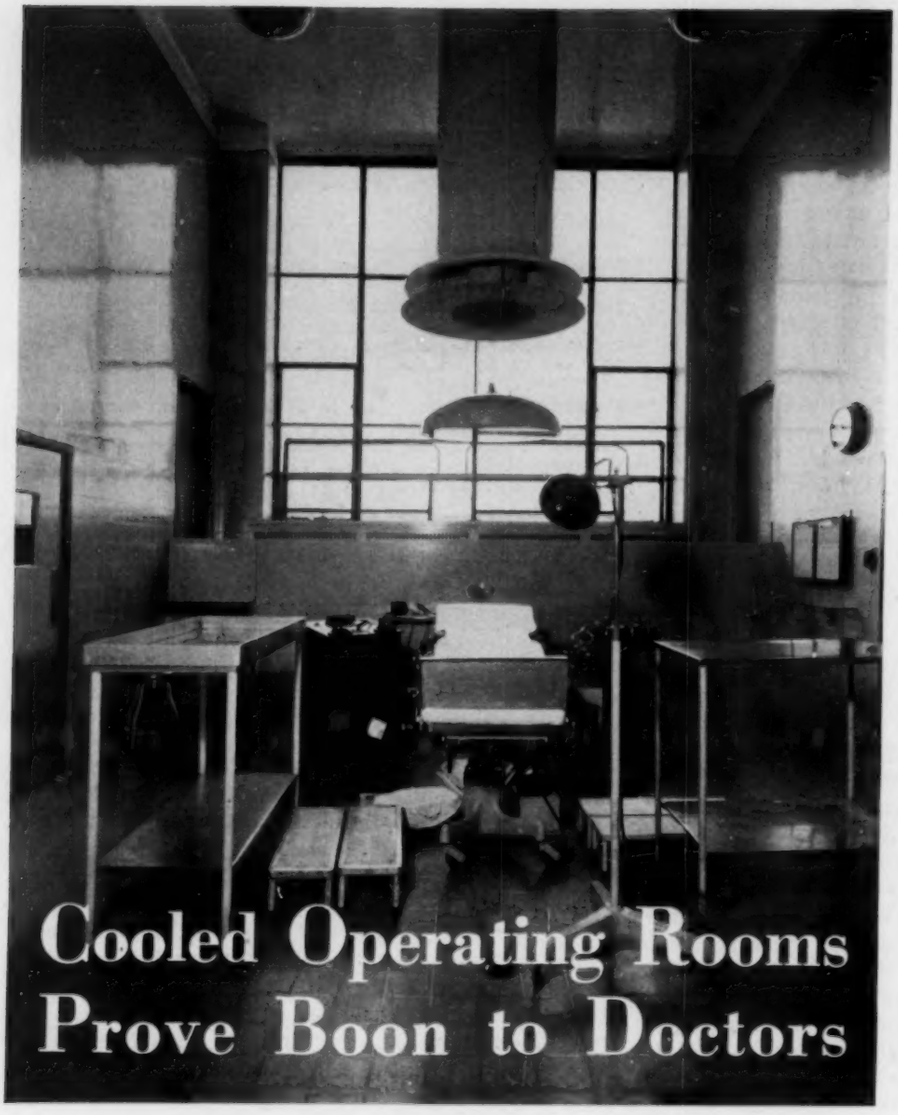
Number	Subject
B16a-1928	Cast Iron Pipe Flanges and Flanged Fittings for Maximum W.S.P. of 125 Lb
B16b-1928	Cast Iron Pipe Flanges and Flanged Fittings for Maximum W.S.P. of 250 Lb
B16b1-1931	Cast Iron Pipe Flanges and Flanged Fittings for 800 Lb Hydraulic Pressure
B16b2-1931	Cast Iron Pipe Flanges and Flanged Fittings for Maximum W.S.P. of 25 Lb
B16c-1927	Malleable Iron Screwed Fittings for Maximum W.S.P. of 150 Lb
B16d-1927	Cast Iron Screwed Fittings for Maximum W.S.P. of 125 and 250 Lb
B16e-1932	Steel Flanged Fittings and Companion Flanges
B16e1-1935	Addendum to American Standard Steel Flanged Fittings and Companion Flanges
B16e2-1936	Pipe Plugs of Cast Iron, Malleable Iron, Cast Steel or Forged Steel
B16g-1929	Cast Iron Long Turn Sprinkler Fittings
B16g1-1937	Addendum to American Standard for Cast Iron Long Turn Sprinkler Fittings

The Unfinished Business of Standardization

At first thought it would seem that when the major products of an industry have been standardized the work would be completed and standardizing bodies such as the M.S.S. could be disbanded, but this is not the case. In the valve and fittings industry much standardization work still remains to be done and in fact the work can never be completed, due to constantly changing conditions. Standards, of course, stabilize products to a certain degree but experience has shown that demands for changes are constantly coming up and have to be considered. New materials, new service conditions, new designs, new requirements of regulatory bodies and many other factors keep the situation in a constant state of flux, and the society must always keep in close contact with the new developments. In connection with the defensive work of the society it must also be remembered that for some unaccountable reason, valves and fittings appear to be very popular as grist for the mills of miscellaneous specification writers, much to the embarrassment of the industry. The special requirements contained in such specifications, representing the ideas of these different individuals, do not permit manufacturers to follow standard production procedure and compel the introduction of job shop methods, with their attendant increase in manufacturing costs.

From this it is evident that both the constructive and defensive work of the society is of a continuing character and can never be abandoned even when most of the important work would appear to have been entirely finished.

The cooperation of all interested parties is solicited in this important development, to the end that (a) the standards put into effect may be sufficiently flexible to avoid serious interference with new developments and general progress, and (b) to avoid the adoption of ill advised mandatory requirements that would result in wasteful economic losses due to premature obsolescence of expensive production equipment.



Cooled Operating Rooms Prove Boon to Doctors

By Lyman R. Flook*

Because the necessary refrigeration capacity was already available, it was possible to convert an existing ventilating system to complete cooling for three operating rooms and a student amphitheater at Billings hospital at a very reasonable cost—less than 80 cents an hour for operation and maintenance. . . . A feature is the special diffusing head for supplying and exhausting air, which is described in detail here. . . . The cooling has proved a boon to the doctors; before, they often complained of severe fatigue and the inconvenience of heavy perspiration which required an extra nurse and the handicap of interruptions to mop faces

CONVERSION of the existing ventilating system for the operating rooms at the Billings hospital of the University of Chicago into a complete air conditioning installation has proved a boon to the doctors and costs less than 80 cents an hour to operate and maintain, as sufficient refrigeration capacity was already available. Our experience during the two years since the work was done is analyzed here, and a cost summary is given.

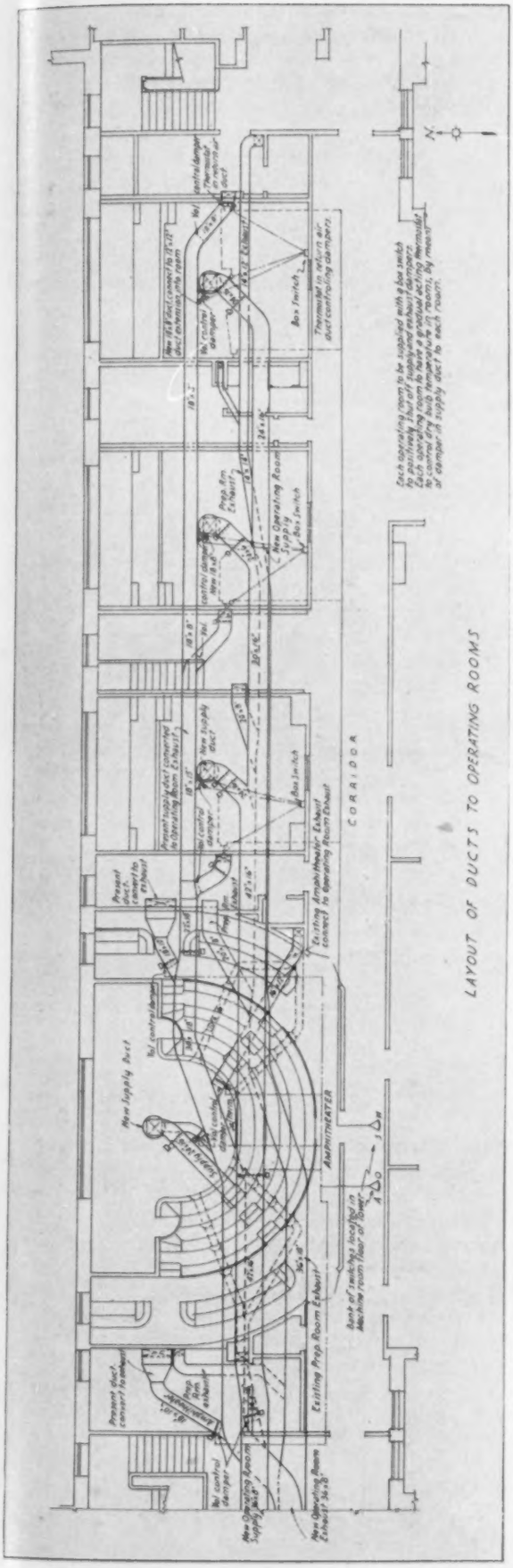
The three operating rooms are on the sixth and top floor of the building, and are arranged in a row on the north side of the corridor, each with its sterilizing, scrub up, and anaesthetizing rooms; they adjoin a student amphitheater with a seating capacity of 110. Formerly, they were ventilated by a large fan equipped with an air washer to supply humidity in winter. Even with flooded eliminator plates, however, the washer failed to remove all of the fine dust. This scheme, with a separate exhaust fan, gave little or no cooling effect, and the doctors often complained during the summer months of the severe fatigue and the great inconvenience of heavy perspiration which required an extra nurse and the handicap of interruptions for mopping faces.

Because the operating rooms work on a busy schedule from 8 a. m. until noon, and the amphitheater is used almost entirely after noon, the duct layout for the modernized system was arranged so that one area or the other can be cooled; the volume of air needed is about the same in each case. The existing supply fan was speeded up to increase its capacity to 6300 cfm to furnish 2100 cfm to each operating room. As each room volume is 7980 cu ft, this represents about 16 air changes per hour. The exhaust fan, with a capacity of 3600 cfm, removes 1200 cfm from each room, the excess of air supplied escaping to adjoining rooms and corridors in order to balance pressures and allow for frequent opening of doors. No air is recirculated. The estimated refrigeration load is 20 tons maximum, 15 tons average.

The outside air intake was provided with a new continuous blanket oil type air filter, and the original washer spray nozzles were changed to an atomizing type; there are two banks each with 16 spray heads.

Chilled water for the sprays is supplied by a brine tank 9 in. in diameter and 100 in. long on the eighth floor level in a tower, to which insulated 2 in. brine lines were run from the existing refrigeration plant in the sub-basement through an elevator shaft. The water flows through a coil in this tank and is pumped to the spray heads by an 85 gpm pump. There is a 150 gal reservoir in the chilled water circuit, and water temperature is controlled by a pair of valves, one of which prevents freezing. A direct brine spray was first used, but because of odors from the operating room —largely ether—and from the brine, the closed circuit interchanger was installed.

*Superintendent of Buildings and Grounds, The University of Chicago.



The air supply ducts run in an attic space under a lead covered concrete roof with 2 in. of fiber insulation. To reduce the temperature rise in this space a small fan, set to operate when the temperature reaches 80 F, was installed to draw outside air through the attic. Despite this fan, however, the temperature sometimes rises to 90 F, causing the temperature of the air in the uninsulated ducts to increase 5 to 6 F. Plans are now being made, therefore, to insulate these ducts.

The cooling system is controlled by pneumatic valves, dampers and adjustable thermostats for fully automatic operation. Switches for on and off control of the supply air are provided in each operating room, which arrangement led to a difficulty that has been met by a small bypass duct and damper at the supply fan to regulate the supply by pressure control, depending upon the number of operating rooms calling for air.

Design conditions provided for maintaining a relative humidity of 55 per cent, and a 16 F differential between inside and outside temperatures during maximum summer conditions. During autumn, winter and spring, room temperatures are maintained (without cooling) at 74 F and 57 per cent relative humidity.

Special Fixture for Supply and Exhaust Air

An original feature developed for this installation is the diffusing head used to supply and exhaust air for each of the operating rooms. Because of the great room height (19 ft, 4 in.), economy required that the cool air be introduced at a considerable distance below the ceiling and below the heat of the lighting fixtures. The arrangement is well illustrated in the accompanying photograph on the next page.

It comprises a double cylindrical duct extending from the middle of the ceiling, terminating in an enlarged diffusing head. Cooled air is introduced to the room through the outer cylinder from the enlarged bottom of the circular diffusion head, which is 10 ft above the floor and directly over the operating table; this gives positive air circulation in the room outward, downward, and upward. The air velocity in the duct is 600 fpm, and the velocity of inlet around the periphery of the head is 150 fpm. Exhaust air is removed through the center of this double duct cylinder.

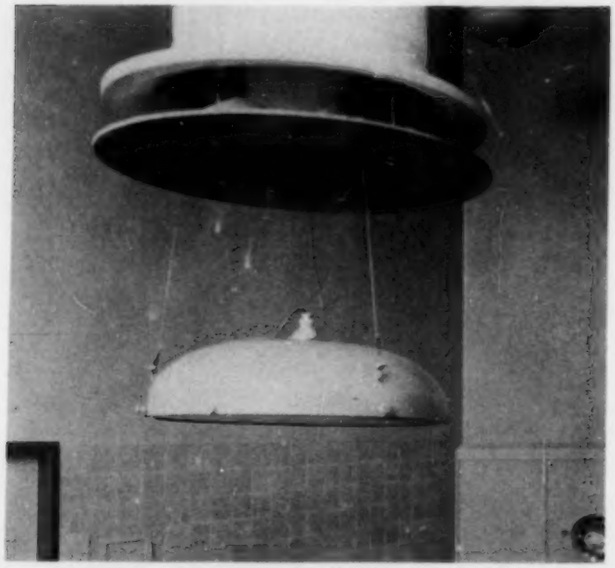
Location of the diffuser head directly over the operating table required that the special "scyalitic" lighting fixture over the table be suspended below the diffuser head and be fully adjustable. To accomplish this, the three cables supporting the light were carried up through the ducts to pulleys and counterweights in the attic.

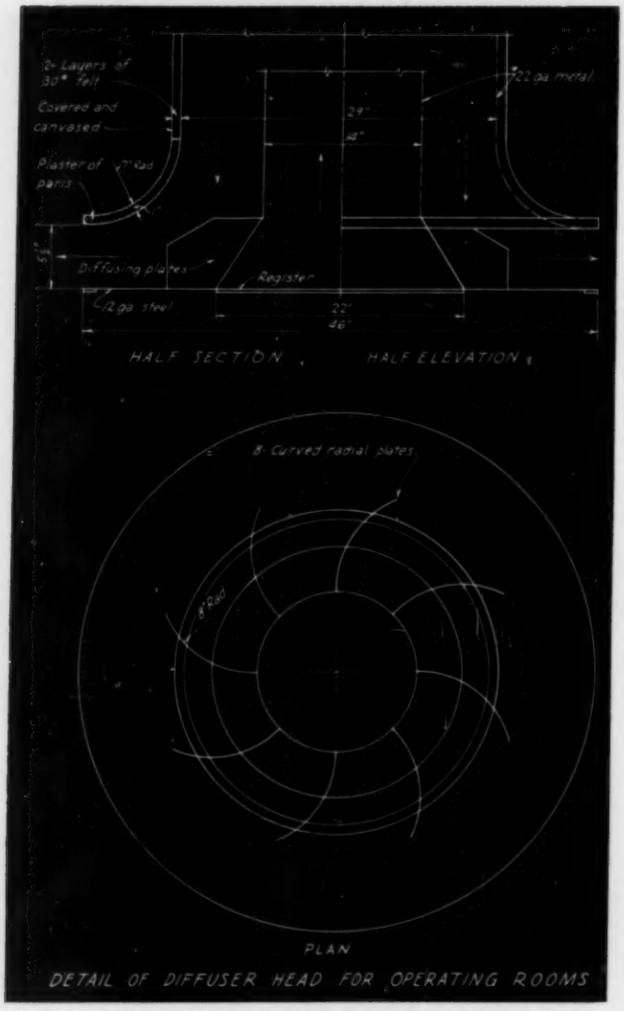
When the drawing for the diffuser heads reached the sheet metal shop where they were to be fabricated, the foreman asked the draftsman just how the bottom—a double curved surface, an arc of revolution—was to be fabricated, and it seemed to be a real difficulty. The problem was solved by having this shape made of plaster of paris by an architectural modelling firm, in accordance with the sketch shown on the next page.

Operating Results and Costs

As mentioned, the installation was designed to maintain a maximum spread of 16 F between inside and

outside temperatures in summer. Operating results have indicated that with relative humidities of 50 to 55 per cent, a spread of 12 F is better practice, as greater differences give too much contrast with the conditions in the hospital wards and lead to criticisms.





DETAIL OF DIFFUSER HEAD FOR OPERATING ROOMS

Records for the extremely warm summer of 1936 when the air conditioning system for these three operating rooms was in full operation compared with the same months of the summer of 1934 (before the system was installed) give the excess hours of operation of the two 30 ton carbon dioxide compressors (which handle all the refrigeration requirements of the hospital):

	1934	1936	Excess
June	527 hr	632 hr	105 hr
July	720	1056	336
August	735	1244	509
September	608	1023	420
	2585	3955	1370

Added costs for materials for these four months were:

	CO_2	Oil	Power	Labor	Total
June	$ 2.00	$ 0.86	$ 91.00	$11.76	$ 105.62
July	7.00	2.69	202.54	48.16	260.39
August	2.50	4.16	300.30	74.48	381.44
September	2.00	3.36	263.90	70.00	339.26
	$13.50	$11.07	$857.74	$204.40	$1086.71

The average cost of $1086.71 for 1370 hours is a unit cost of 79.3c per hr of operation, exclusive of fixed charges. Since reserves for replacements or for amortization are almost unknown in educational accounting practice, no such records are kept. To compare costs with commercial practice, however, estimates of these costs are here made, including interest on the cost of the improvements, amortization on a 10 year expectancy, insurance, taxes, etc.:

6 per cent on $10,000	$ 600	
10 per cent on $10,000	1000	
Insurance, taxes, etc.	100	$1700

Based on 1370 hours of use, the cost would then be

Operating	$1087	$0.79
Fixed charges	1700	1.24
	$2787	$2.03

Because 1936 was an unusually warm summer, the large number of hours of operation used as the divisor should be reduced to perhaps 1200 for Chicago. Using $1000 as the cost of operation for the fewer hours of operation, in this case the hourly cost would be, very closely:

Operating	$1000	$0.833
Fixed charges	1700	1.417
	$2700	$2.250

As above stated, the costs here given are for the particular installation, where the compressor and its brine system were already in operation. This fact is not reflected in the above costs. Had an entirely new plant been purchased, the costs would have been much greater.

Table 1—Observed Dry and Wet Bulb Temperatures (F) and Per Cent Relative Humidities During 1937

Date	Outdoor		Spray Water Temp. F	Air				
					To Room		From Room	
	D. B.	W. B.		From Fan	D. B.	W. B.	D. B.	W. B.
Aug. 10 (9 a. m.)	78	72	44	69	72	66	77	68
Aug. 11 (3 p. m.)	88	75	42	69	73	66	78	67
Aug. 14 (2:15 p. m.)	85	66	42	66	74	66	77	67
Aug. 16 (2:30 p. m.)	94	68	41	69	74	64	76	65

E. Vernon Hill's* Page

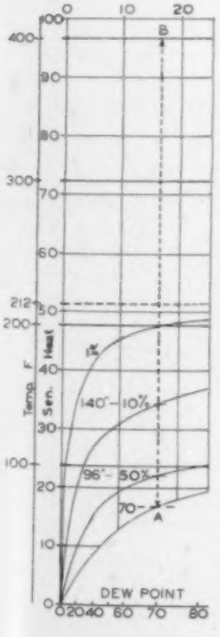
Saturated Air Above 212 F

I PROMISED last month to discuss in more detail the subject of saturated air above 212 F. Before going into the subject, however, it is perhaps well to give definitions for relative humidity and saturated air. We find by consulting the 1937 A.S.H.V.E. "Guide", Chapter 1, that *relative humidity* is defined either as the ratio of the actual partial pressure of the water vapor in air to the saturation pressure at the dry bulb temperature, or the ratio of the actual density of the vapor to the density of the saturated vapor at the dry bulb temperature. The latter is the correct definition of relative humidity, instead of the definition so frequently used, "the amount of moisture in the air expressed as a percentage of the amount it can hold when saturated at the same temperature".

For a definition of saturated air, we refer to Chapter 44 in the "Guide", where we find:—"Saturated Air. Air containing as much water vapor as it can hold without any condensing out". The definition continues "The partial pressure of the water vapor is equal to the vapor pressure of water at the existing temperature."

The first part of this definition is misleading. It is a concession, no doubt, to a well established nomenclature which conceives of air as acting like a sponge, absorbing water vapor. The latter part of the definition is explanatory and correct.

We must bear in mind, however, that saturated air (so called) is nothing more nor less than a mixture of air with saturated vapor and the proportions in the mixture depend upon the temperature. Saturated air, for example, at a temperature of 100 F is a mixture of saturated vapor at a vapor pressure of 1.931 in. of mercury with a sufficient volume of air to exert a pressure sufficient to make up the difference between 29.92 in. and 1.931 in., or 27.989 in. of mercury. In other words, if we have a container holding exactly 1 cu ft of dry air and we wish to saturate it, we force into the container a certain amount of water vapor until the pressure of the vapor is 1.931 in. of mercury. In doing this, we *force out* some of the air having a total pressure of 29.92 in. until the remaining volume of the air exerts a pressure of 27.989 in., so that the total of the partial pressures of the mixture will equal the total pressure of 29.92 in.



The air does not absorb moisture and must be removed from the container to accommodate the water vapor; the water vapor *only* can properly be said to be saturated.

Now let us refer to last month's problem of the furnace with the evaporator pan in the bonnet. To understand this clearly, we will construct a psychrometric chart in accordance with customary procedure, except that we will design the chart on a heat unit basis. We will plot sensible heat against latent heat. Such a chart is shown here. For ordinary purposes such a chart does not extend above 100 F dry bulb, which is 24 Btu sensible heat; it need not extend to the right more than 25 Btu latent heat, which would be about 82 F dew point. We will extend the chart upward, however, to 100 Btu sensible heat to be in the temperature range above 400 F.

In the definition of relative humidity, it was stated that relative humidity is the ratio of the actual density of the vapor to the density of the saturated vapor at the dry bulb temperature. As the vapor in the air mixture determines the latent heat, we may also determine the relative humidity by finding the ratio of latent heat in the air vapor mixture to the latent heat at saturation.

The curves at the lower part of the chart are plotted as percentages of latent heat at saturation and they are continued upward until they leave the chart at the right border. Note that these latent heat curves which are also relative humidity curves, are plotted for 50, 10 and 1 per cent, and that the 1 per cent curve flattens out and disappears to the right slightly above a temperature of 200 F. It is apparent therefore, that air, from any point on the chart, raised above 200 F will be above or to the left of the 1 per cent curve and will have a relative humidity of less than 1 per cent.

Take for example, air at point *A* on the saturation curve. This is at a temperature of approximately 70 F and a relative humidity of 100 per cent. If air under these conditions is heated, it would follow the vertical line *AB*. As we pass 96 F, the air would have less than 50 per cent relative humidity; as we pass 140 F, the relative humidity would be reduced to less than 10 per cent and as we pass 200 F, the relative humidity would be less than 1 per cent. If this air circulates through the ducts of a furnace heating system having a relative humidity of less than 1 per cent and is discharged into the room and cooled to 70 F, it will again be saturated air, very uncomfortable and possibly be the cause of considerable damage. As a matter of fact, we cannot discharge air with this high humidity with safety. This analysis explains the reason the statement was made to the furnace man that air at 400 F could not possibly carry more than 1 per cent relative humidity.

The argument hinges of course, on what meaning we ascribe to the term relative humidity above 212 F. We cannot apply the definition of relative densities of the moisture in the air to the moisture at saturation, because we cannot have moisture at saturation above 212 F at atmospheric pressure. The only reasonable definition that has any meaning whatever is the percentage of moisture in a given mixture of air and water vapor when

[Concluded on page 189]

*Director of Research and Education, Refrigeration and Air Conditioning Institute. Member of Board of Consulting and Contributing Editors.

Piping for Indirect Refrigeration

AT least 80 per cent of all refrigeration work is done by direct refrigeration, in which case the refrigerant is applied in a container, cooler or evaporator directly to the material to be cooled. In the remaining cases, it is done by air, water, or brine which acts as an intermediary or heat conveyer, although the final cooling is done by a direct expansion refrigerant. Hence, these systems are called "indirect refrigeration".

It is the purpose of this article to consider the piping for indirect refrigeration systems using water or brines.

Advantages of Indirect Refrigeration Systems

There are many advantages in using a medium such as water or brine for indirect refrigeration (water being used for relatively high temperature work and brine for low temperature work):

(1) In some industrial processes, use of ammonia, for example, is impossible on account of the possibility of leaks damaging the product.

(2) In the long term storage of foods in warehouses, the indirect system is not only safe and convenient but also affords split degree temperature control.

(3) To meet sudden peak demands, it affords a means of storing refrigeration during periods of low demand by using a compressor of low average capacity over a long period of operation.

(4) It is often used as a spray wherever high humidities are also wanted, as in the case of packing house chill rooms, where the air is washed, cooled, and humidified to avoid meat shrinkage.

(5) It is mandatory for glazing fish.

(6) It is used extensively for the production of frozen foods.

(7) It is necessary for "freezing needles" used in freezing earth to prevent earth slides at excavations, dams, etc.

(8) It is used in congealing and holdover tanks and more recently in eutectic plates where different strengths are used according to the temperature desired.

(9) It is used extensively in large markets and multiple systems of widespread layout or in a multi-floor arrangement. In fact, the codes of some cities will not permit direct refrigeration for industrial applications above the first floor.

(10) In refrigeration distribution systems in hospitals, asylums and similar institutions, brine is required, for the patients are confined and unable to escape or avoid a possible gas leak.

(11) The piping distribution system carries a very low liquid pressure and all higher pressure gas lines are situated centrally in a remote machine room.

(12) If a leak does occur, it does not interfere with business in a commercial installation or have any pathological effect on nearby people.

Disadvantages of Indirect Refrigeration Systems

Notwithstanding the advantages of the indirect system, it has some disadvantages:

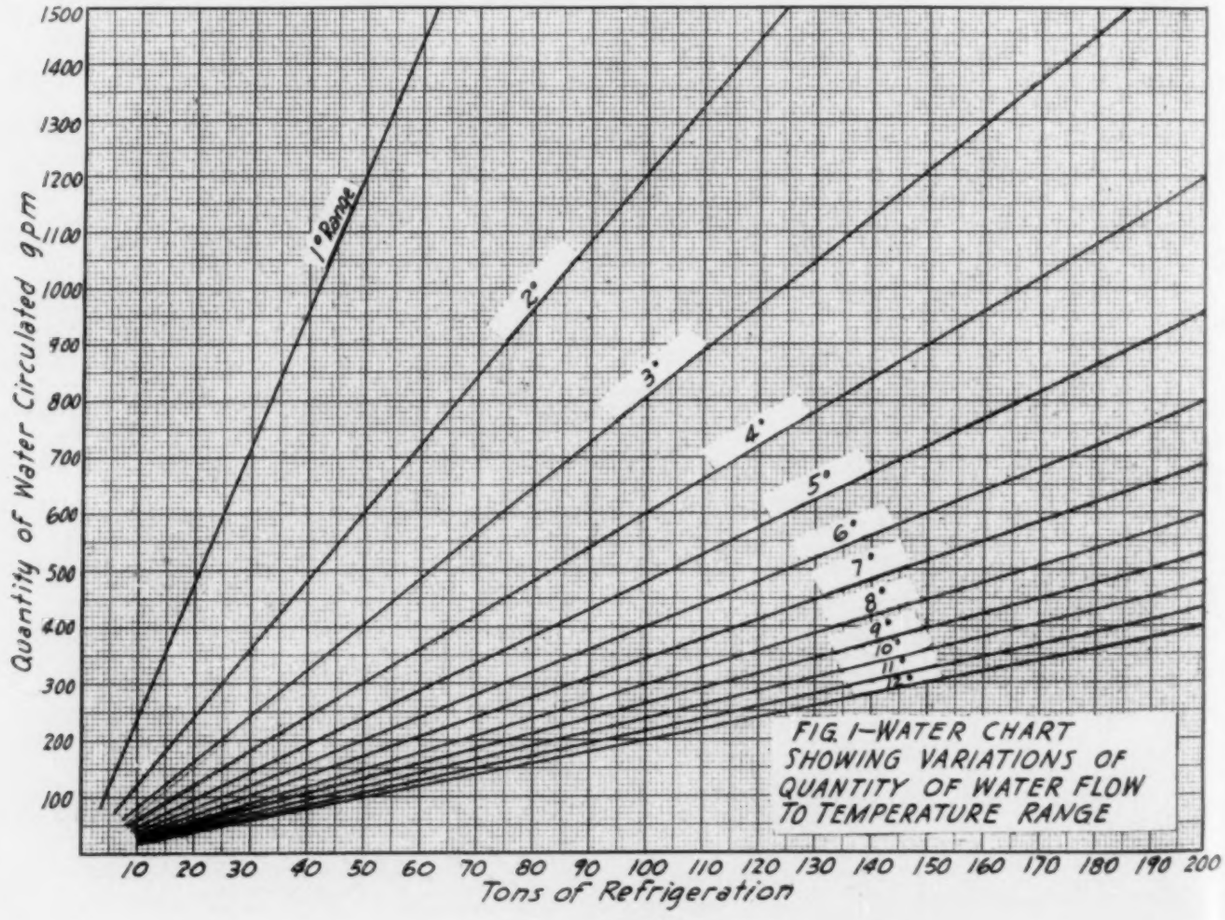


FIG. 1—WATER CHART SHOWING VARIATIONS OF QUANTITY OF WATER FLOW TO TEMPERATURE RANGE

R. C. Doremus* gives information for selecting the right pipe sizes for indirect refrigeration systems using water and calcium chloride brine as the heat conveying mediums; his next article will present similar data for sodium chloride brine. . . . These articles complete those which have appeared in recent issues of H. P. & A. C. on the design of piping for direct refrigeration using various refrigerants

(1) Its first cost is more than a direct refrigeration system.

(2) It is more expensive to operate than the direct system on account of requiring two temperature differences—(a) material to brine, and (b) brine to ammonia.

(3) It requires a higher compressor displacement for any given tonnage on account of requiring a lower evaporator pressure.

(4) It causes a higher compressor bill for power for the same reason.

(5) The power bill is further increased by the power to drive brine circulating pumps.

(6) It requires larger distribution mains for slower fluid velocity which means higher cost for pipe, valves, fittings and pipe insulation.

(7) It has a higher corrosion loss and requires more attention in the proper use of corrosion retardants.

(8) The use of dissimilar metals must be avoided in the piping system to prevent electrolysis or galvanic effects.

(9) It requires accurate pH control, a subject still not too well understood by operating engineers.

(10) The medium is liable to freeze solid and split open a part of the equipment either through being too weak in gravity or by too cold an evaporator temperature.

It is evident that these advantages and disadvantages of the indirect refrigeration system must be carefully considered before an intelligent decision can be made.

Water as an Indirect Refrigerant

In many cases, water may be circulated as an indirect refrigerant provided the temperature of the supply is above the freezing point. In common practice, it is customary to use a supply temperature of 34 F or above in order to allow a small margin for safety.

The quantity of water circulated will depend on two factors: (1) The tonnage capacity. (2) The range in temperature or rise permissible between supply and return. In connection with water circulating systems, it is well to remember a little rule of thumb. The product of the gallons per minute per ton of refrigeration and the temperature difference between supply and return will always be a constant 24.

Thus, if we circulate 4 gpm per ton, t_2-t_1 will be 6 F.
And, if we circulate 3 gpm per ton, t_2-t_1 will be 8 F.
Also, if we circulate 2 gpm per ton, t_2-t_1 will be 12 F.
Also, if we circulate 1 gpm per ton, t_2-t_1 will be 24 F.

Fig. 1 shows graphically the quantity of water that must be circulated in gpm for any tonnage of refrigeration for any desired temperature range or rise. Obviously, if the amount of water circulated is large, the temperature range will be small, and one might conclude that the greater the amount of water circulated, the better. However, we all know there are practical limitations in pipe costs and cost of power for pumping. If we are computing an air conditioning job, we will probably want

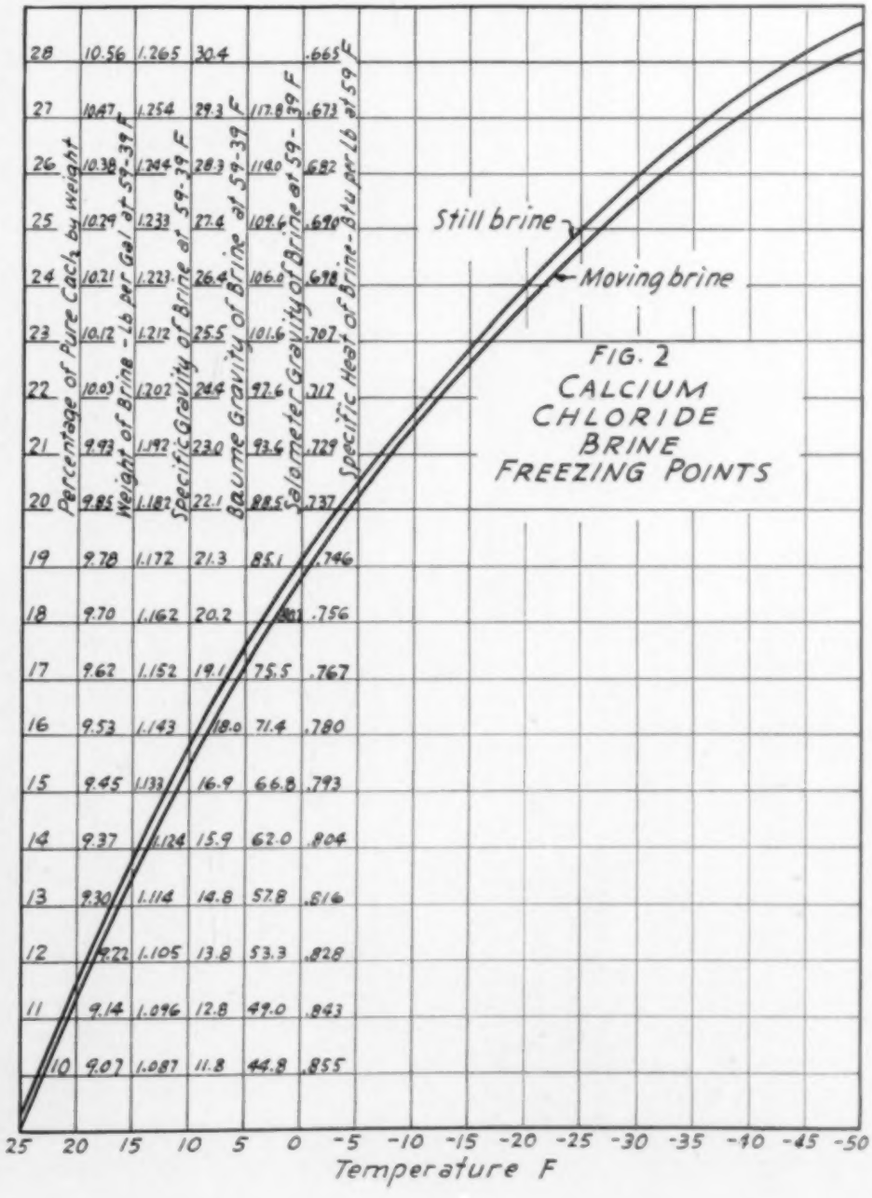


FIG. 2
CALCIUM CHLORIDE BRINE FREEZING POINTS

*Chief Engineer, Detroit Ice Machine Co. Member of Board of Consulting and Contributing Editors.

to circulate about 3 gpm per ton so that the range of water temperature will be about 8 F. In this case, we will probably use a finned coil or similar surface and we will want to work with a mean temperature difference between air and water that will be as large as possible in order for that surface to be economical in cost.

In case we are not limited by surface cost, the other practical limitation is the cost of piping. The most desirable size pipe to use for circulating water may be determined from a water pipe friction chart showing pressure drops and velocities. Velocities from 5 to 8 fps cause low friction drops and will therefore permit low pumping costs.

Calcium Chloride Brine

In case the fluid supply temperature must be below 34 F, it is too cold to use water and a salt must be added to lower the freezing point, thus forming a brine. The most commonly used brine for this purpose is made from calcium chloride ($CaCl_2$). Commercial calcium chloride for refrigeration is from 73 to 78 per cent pure. Large buyers obtain it in tank car lots in liquid form while smaller buyers or those not having railroad sidings obtain it in solid, flake or granulated form.

The strength of calcium chloride brine that must be maintained for proper circulation is naturally determined by the brine temperature desired, and its freezing point. Usually the brine supply temperature must be from 10 to 20 F below the coldest item to be refrigerated on the system, and the ammonia temperature used to cool the brine must be from 10 to 15 F below the brine. If we want to freeze the brine in a congealing tank, it must have a freezing point about two degrees above the ammonia evaporator temperature. If, however, we do not want to freeze the brine but desire to keep it in circulation, it must have a freezing point below the ammonia evaporator temperature, and for a safe margin, this should be about three degrees or more.

Fig. 2 shows the gravities and freezing points of calcium chloride brines, from which the proper brine strength for any system may be determined. If the brine in an existing plant becomes too weak through the addition of water and is liable to freeze, this chart will facilitate the calculation of the quantity of calcium chloride that must be added to give the brine the proper strength and freezing point.

The specific heat of calcium chloride brine varies with the strength and temperature. Average values are given on this chart to assist in making calculations. For extreme accuracy, engineering data tables should be used and these are available carried out to the fourth decimal place.

There is a rule of thumb applying to calcium chloride brine similar to the one given for water. It varies somewhat because the specific heat varies, but for a zero F brine, the gpm per ton multiplied by the temperature range will equal 27.

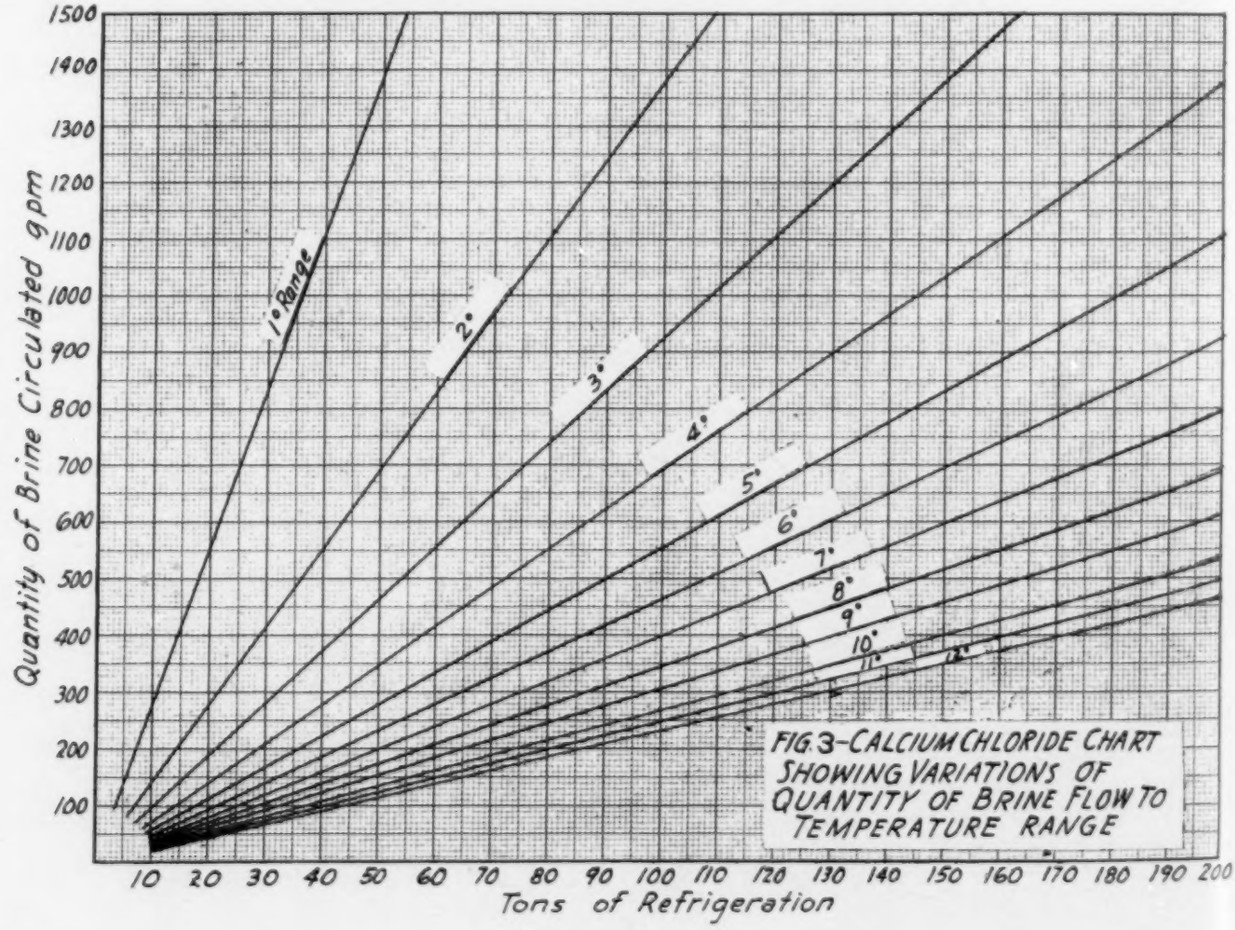
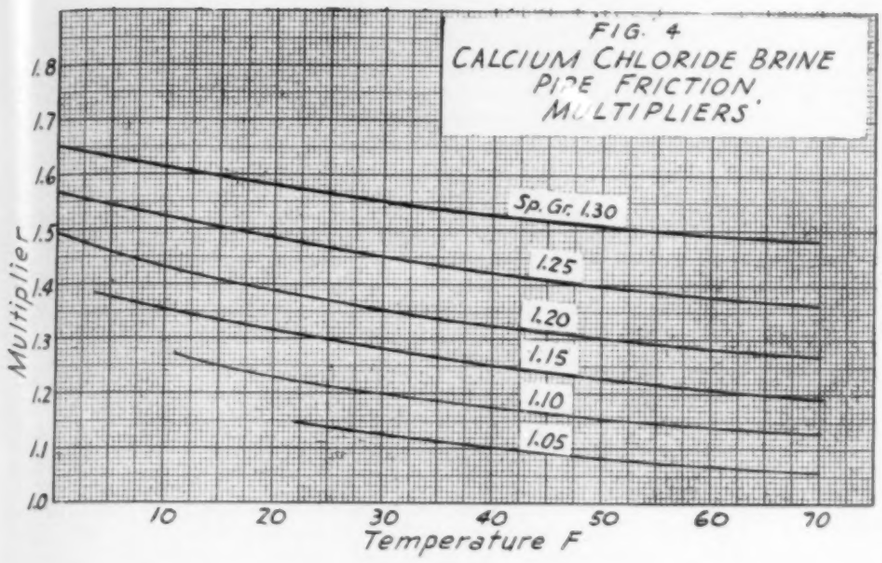


FIG. 3—CALCIUM CHLORIDE CHART SHOWING VARIATIONS OF QUANTITY OF BRINE FLOW TO TEMPERATURE RANGE



Thus, if we circulate 6 gpm per ton, the $t_2—t_1$ will be 4.5 F.
And, if we circulate 3 gpm per ton, the $t_2—t_1$ will be 9 F.
And, if we circulate 2 gpm per ton, the $t_2—t_1$ will be 13.5 F.
And, if we circulate 1 gpm per ton, the $t_2—t_1$ will be 27 F.

Fig. 3 shows the quantity of calcium chloride brine in gpm that must be circulated for any refrigeration tonnage to maintain certain temperature ranges between brine supply and return temperatures. For an average condition, this is based on a brine supply of 15 F and an ammonia temperature of zero F. The strength is a specific gravity of 1.180 for about—2.5 F freezing point, at which point it has a specific heat of 0.738. For other brines, calculations should be made to obtain definite quantities.

For determining proper pipe sizes, pipe friction and pump head specifications, a pipe friction chart should be used. Fig. 4 shows a few curves from which a multiplier can be found. First, find what the pipe friction would be if the fluid were water and then multiply this by the calcium chloride friction multiplier to determine the actual pipe friction when using calcium chloride brine instead of water. The pipe size should be chosen to give a fluid velocity of from 4 to 7 fps for economical friction heads.

Calcium chloride brine is used for low temperature refrigeration work, for its strength may be increased to obtain a freezing point of approximately 53 F below zero. However, before it freezes solid it becomes very viscous and slushy and should not be used for circulation below about —45 F.

[Mr. Doremus' next article will give data on sodium chloride brine, and will discuss piping materials for indirect refrigeration systems, and brine corrosion.]

Largest Wood Pipe

This pipe at Outardes, Quebec, is 5888 ft long, 17 ft 6 in. inside diameter and was built for a maximum head of 120 ft. The staves are $4\frac{11}{16}$ in. thick, pressure creosoted, and have patented double tenon butt joints for joining individual staves. The pipe rests in structural steel cradles which in turn rest on concrete footings. The line is used to carry water for hydro-electric power development in a 70,000 hp plant which furnishes electric power for the new paper mill of the Ontario Paper Co. at Baie Comeau, 14 miles east of the power plant. The entire pipe structure was designed and installed by Canadian Wood Pipe & Tanks Ltd.

E. Vernon Hill—

[*Concluded from page* 185]

heated from ordinary temperatures to the temperature under consideration.

Naturally, we could extend the chart to the right until the saturation curve reaches the dry bulb temperature of 212 F; and we could saturate the air at these higher temperatures, but under such conditions we obtain a mixture in which the percentage of air becomes less and less, until at 212 F the mixture is all vapor and no air. We are dealing in the present instance with air in a furnace heating installation, and so need not consider air saturated at temperatures approaching 212 F.

As a matter of fact, however, it is not good practice—it is not even technically correct—to speak of relative humidity above 212 F.

This argument was started for the purpose of illustrating the absurdity of our every day usage of the terms "saturated air" and "relative humidity" as the amount of moisture air can hold at a certain temperature. In this discussion, I do not wish to convey the impression that more moisture cannot be added to air at high temperatures than the amount indicated. As a matter of fact, any proportional mixture of steam and air may be maintained, but it is strictly a mixture of air and superheated steam, with no relationship that we are able to follow to relative humidity other than the one indicated.

Here's How I'd Air Condition an Office

Readers of H. P. & A. C. Tell How to Air Condition Typical Office Suite Diagrammed in December Issue

HOW best to apply winter and summer air conditioning to a typical office layout involves consideration of a number of practical factors other than equipment design and selection. As the subject has wide applications, readers were invited in the December H. P. & A. C. to answer a question asking for general recommendations on the method of air conditioning a suite of offices. From the replies received, those appearing here have been selected for publication.

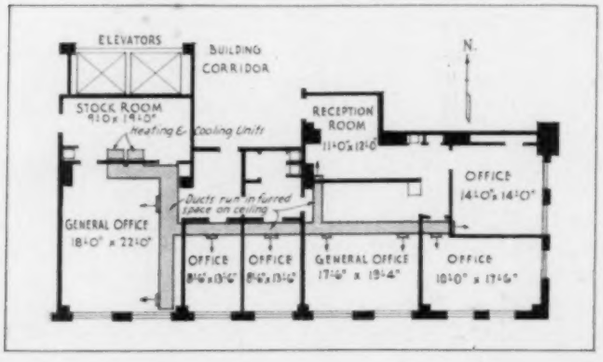
The comments are concluded with those of the manager of the building in which these offices are situated, as he is familiar with the building with which the air conditioning system would have to be coordinated, and has had experience with air conditioning equipment in other buildings he has operated.

Following is the December question:—"We are tenants of a large office building in a middle western city, and are interested in providing our offices with winter air conditioning at least, and possibly year 'round conditioning, for the comfort and efficiency of our employees. What general type of system would be most suitable for us, and what would be its advantages and disadvantages? We do not want a detailed design for a system, but rather general recommendations and helpful comments.

"Our offices comprise the south-east corner of the top floor of the building. The building is heated entirely by direct radiation, vacuum system, and has no air conditioned areas except for stores and shops on the ground floor. Our lease is for a five year period (of which one year has already run) and while we do not contemplate moving when it expires, the fact that we are building tenants and not building owners no doubt affects the matter."

TWO central units complete with a steam coil in each should be placed on the south wall of the stock room in parallel to a duct system which can be furred in neatly on the walls at the ceiling as shown. Grilles of the correct size should be placed on the face of the furred space with a connection to the main, and properly dampered. To return the air to the units, equip all doors with mechanically operated metal louvers, except those leading to other parts of the building. In this way, the recirculated air will find its way back to the units which are partially open to the stock room.

Proposed duct layout for office suite



As these offices are on the top floor, a duct can be run through the ceiling to the roof for the outside air intake. This should be large enough to allow all or a fraction of the total volume of air circulated to be outside air. The system should be thermostatically operated and controlled so that each unit may be operated separately or in conjunction with the other to provide the most efficient operation under changing load conditions.

Steam and water lines can be connected to the units from the building mains. The radiators should be left in place and used only when necessary.

The advantages are greater working efficiency due to better air conditions; comfort for the employees; the units may be disconnected and used in any other office should the tenant move; simplicity of installation; and low maintenance cost. The disadvantages are less space in the stock room, and initial cost of the installation of ducts and registers.—HOMER L. YARYAN, JR., Heating and Ventilating Engineer, Mills, Rhines, Bellman & Nordhoff, Inc., Architects and Engineers.

ASSUMING that direct expansion of "Freon" is to be used for cooling, two possibilities present themselves for this sized job: the central plant, duct distribution system; and individual units for each office.

The load figures about 5 tons on a 15 deg differential, especially if the building management would help to the extent of 4 in. of rock wool on the floor of the attic space over the offices. If they refuse, it would still pay to have the ceiling insulated, thus saving in equipment first cost, motor operation and city water cost.

Outside air for the private offices is not of prime importance, since the occupancy is small and window ventilation can be resorted to in emergency. The general offices and reception room should on the other hand be supplied with at least 20 cfm of outside air per person—preferably 30 cfm if the occupants smoke.

Individual self contained units are expensive in first cost, occupy valuable floor space and are not, in the larger sizes, particularly sightly, especially in the "under-the-window" type necessary in the general offices where outside air connections must be made to the unit. The reception room also presents a rather impossible problem in considering the individual room units on account of the absence of any window to use for outside air supply. The four private offices and even the reception room could be equipped with room units without provision for outside air and the condensate caught in buckets instead of being blown out through the window opening. The advantage of this scheme lies in its relative simplicity of installation, its mobility in case of giving up the offices, and particularly in the excellent control of individual room conditions. It is true, however, that special electric wiring would have to be run to all units and ½ in. water and waste to each of the five smaller units. The two general office units would be air cooled.

A variation of this scheme would consist in the installation of a single condensing unit in the stock room, with liquid and suction piping run to an evaporator unit or room cooler in each room. In this case the piping is costly, unsightly and liable to leaks due to the large number of exposed joints. It would, however, concentrate the noise of the reciprocating condensing unit in one place where it could be dampened if necessary. The total cost would probably be less than for the self contained units, wiring to the units limited to fractional horsepower fans, and machinery complications and consequent service calls considerably lessened. Outside air supply for any such units, however, again presents a problem unless the units were specially built. Water condensed from the larger units would have to be piped away.

The most practical method to use in this case is undoubtedly

the central plant and duct distribution system, although a considerable concession must be made to lack of control in the various quarters. In the morning, even with the east and south windows well shaded by awnings, the reception room and east private office will be probably overcooled. Good control of temperature may be had, it is true, by automatic volume dampers in the supply outlets actuated by a thermostat in each room, but only at the expense of ventilation requirements. Experience shows that this is a dangerous procedure. The occupants are of necessity instructed to keep the windows closed and with the air limited or cut off entirely in any one room, the difference in the condition of the air in the room is quickly felt.

Constant air supply and perfect control may be obtained, of course, by the rather expensive method of a mixing chamber at the central unit, with individual ducts leading to each conditioned room and a thermostat in each room operating double mixing dampers in the plenum chambers. For all practical purposes it is probably as well to admit that there will be some variation in the condition of the various rooms and to try to average this out by a well considered location of a single thermostat—preferably in the path of the recirculated air from all of the rooms.

In this case the conditioning unit, containing renewable or throw away filters, direct expansion coil, steam coil, humidifier, fan and motor, would be hung on the ceiling of the stock room against the partition wall between the two doors leading to the large general office, with the condensing unit located directly under or in any other practical place in the room. Possibly the elevator penthouse directly above could be used to house the condensing unit, if it is not more than 12 or 15 ft above the evaporator coil. This would prevent possible objectionable noise. The conditioning unit might also be located in the penthouse, the discharge duct run down through the ceiling of the stock room, and the recirculating air duct up through the stock room ceiling. This would be rather expensive however and not as practical as locating the conditioning unit on the stock room ceiling.

In the latter case, the recirculated air duct (about 30 x 8 in.) could be run directly up to the unit from a large grille in the wall of the general office between the doors leading to the stock room. The outside air might well be brought down from the penthouse on one side of the elevator shaft as it would only require a duct approximately 24 x 6 in. This would avoid entirely the necessity of cutting through the roof. The recirculated air can be taken from the various offices through door or wall grilles near the floor, into the large general office through the small hallway, without causing objectionable drafts.

The fan discharge air can of course be carried through flat galvanized ducts on the ceiling of the small hallway, through the small general office close against the reception room partition and connection made into each office through deflecting type grilles high up near the ceiling. The grilles should preferably have a volume damper in the duct branch, or be louver damper registers, so that the room occupant may operate them from the floor with a stick or chains. This is probably more psychological than useful as it has been found that the average person will stand a considerable variation in temperature without discomfort or notice—unless he happens to be "thermometer conscious."—J. Clifford Lewis, Lewis Engineering, Inc., Consultants.

IT seems that the tenant is concerned with the following problems:

1. The benefits of winter air conditioning are desired from a health standpoint to keep down colds and to increase the health and efficiency of his employees in general. Of course, this is accomplished by controlling temperature and humidity, providing outside air for ventilation, and filtering all of the supplied air.

2. Low first cost must be considered because of a definite lease on the office space. However, this does not mean sacrificing any item of the system that will affect the final results, as the primary object is to provide healthful conditions.

3. Since the tenant does not know whether he will install summer air conditioning on the initial installation, a system should be designed so that it can be converted to summer cooling without alterations to the original winter air conditioning system.

4. A choice must be made between a central station unit with a duct system or a number of individual units for the various offices. The first cost of individual units would most likely be higher. As refrigeration equipment is not contemplated immediately, the unit conditioner would not be in the picture due to the cost of installing refrigeration at a later date. Although the salvage value of the individual units might be slightly greater than central station equipment, it is doubtful that this item will be worthy of consideration because of higher first cost of the individual units. Then, too, as the individual unit models are changed yearly, it depreciates more rapidly than the central station equipment, which is almost unchanged from year to year. For this installation a central station job would be recommended.

As a year 'round job is contemplated, the ducts should be sized for summer cooling and zone control should be considered. In this particular problem, the two private offices facing east are the only ones which might require zoning.

There seem to be two possibilities of installing the duct work. One method would be to run the ducts exposed through the offices and the other to install the ducts in the attic space and discharge the air through ceiling plaques. The first method will be the cheapest, but the second method will make a neater and more attractive installation. When the tenant installs summer air conditioning, the attic duct system will have to be insulated. However, as it will prove economical to insulate the roof to cut down the refrigeration load, it is possible to place the roof insulation above the ducts and thus eliminate the cost of duct insulation.

For this job a standard factory built air conditioning unit would be the most economical and practical to use. This should contain a fan, heating coil, face and bypass dampers, filters and humidifier with a space for a cooling coil arranged so that the cooling coil can be added at any time. The face and bypass dampers are recommended because this will allow the use

How would you air condition an office in one of these buildings? Other readers give their ideas here

of an "on and off" steam valve which will eliminate the need of a preheater. This unit could be either in the stock room or attic space.

Other general recommendations would be to take return air from the floor level in the two general offices; take outside air from the roof or attic and size the outside air duct to handle 100 per cent capacity of the fan so that the fan can be used to cool in intermediate seasons and ventilate during summer, and install relief grilles in the doors of all private offices to insure a definite air circulation.

If the present heating system is in good condition a "split system" would be recommended; that is, allow the present radiators to take care of heat transmission loss and supply heat required for the ventilation air by the conditioning unit. A volume damper in the supply duct to reduce the amount of air during the heating season is desirable to reduce complaints caused by air motion or drafts.

The automatic controls should be simple. A return air thermostat and humidistat will be sufficient to give good results.

In regard to refrigeration equipment, a direct expansion system using "Freon" will be the most economical. The type of condensing equipment (such as a water cooled condenser, cooling tower or evaporative condenser) is a local problem for the tenant and application engineer to decide. This will be determined by the cost of water supply, etc.—W. E. Cone, Engineer, Shook & Fletcher Supply Co.

The Building Manager's Comment

IT is interesting to note from a majority of the ideas submitted that there is a tendency toward the central duct system, and I believe after investigation that this is logical for an office the size of which was shown on the sketch submitted.

There are a few conditions in most existing office buildings which I believe should be brought to the attention of air conditioning engineers and contractors. I believe that if the air conditioning industry will become better acquainted with existing conditions in most office buildings, it will be much easier to avoid mistakes and misunderstandings in the installation of the equipment.

Taking this office layout, let us assume that the compressor and the conditioning unit will be installed in the stock room, which is the logical location. The only floor space lost is that used for the compressor, as the conditioning unit can be set on an insulated frame directly above the compressor. In this connection, I would like to bring out the point that it is not always practical, and oftentimes the result has been unsatisfactory, when the unit is suspended from the ceiling, as oftentimes vibration takes place and is transmitted to the floor above with the result that the occupant of that space complains bitterly. By using the frame to set the conditioning unit on, it is much easier to insulate in two or three places, and to eliminate possibility of transmitting noise to the building structure.

In most offices of the type under consideration, there will be a stock room or a storage room available for the installation of equipment, and in most cases, the stock room will be reasonably close to a wet stack, as is the case in this layout. Therefore, water lines and waste lines are easily available. Also, the stock room, or the less valuable space in any office, is that nearest the building corridor, and this is advantageous, as will be brought out.

There is a very important point to be considered in connection with year around air conditioning, and that is in regard to the steam supply for the heating coils. Let us take a typical office building and analyze the heating system. The system was installed to care for direct radiation and the risers, pumps and boilers were sized accordingly. Now let us assume that a tenant desires year around air conditioning and the coils are connected with the existing steam risers. Perhaps this one installation will do no particular harm, although if it is large enough and the riser is not amply sized to start with, one installation may even cause trouble. Even though the one job does not cause trouble, if many more jobs are installed and connected to the existing system, it is seen that the steam piping and the boiler room equipment may be overloaded and, at the same time, in order to supply steam to the blast coils in the air conditioning equipment, it will be necessary to keep much more steam on the balance of the building than is necessary for comfortable heating, with the result that the majority of the building will be overheated with resultant waste of fuel. Of course, the latter objection would be eliminated if the air conditioning equipment were located close to the boiler room, in which case it is a simple matter to run a direct steam line from the boiler header back of the pressure reducing valve, in case there are pressure reducing valves on the job. However, most installations of air conditioning equipment are a considerable distance from the boiler room and the running of special steam lines is not always practical.

The question, therefore, is what is the next best thing. Consider the amount and movement of air through the main corridors of the building. It is known that there is a strong updraft in any building of any size, and we know that in order to have this updraft, there is bound to be a considerable amount of air taken into the building at its base. We have found from taking readings with recording thermometers that the temperature of this air is very uniform, and we have found from readings taken in various buildings that the temperature in winter will stay very close to 70 F and does not vary much over 1 F during a day.

In view of the fact that it is not always practical to connect to the existing steam lines, and considering the uniform temperature and the amount of outside air coming into the building corridors, I see no reason why the air supply cannot be taken from the building corridors. In fact, it has been done and has been proved to be satisfactory in cases with which I am familiar. One advantage of this idea is that much more outside air can be brought into the room than if it were brought in directly from the outside. The next step of course, is to provide openings for exfiltration, and inasmuch as the room is under pressure, the simplest way is for a grille to be installed in a window, or windows to be partially opened to allow the escape of the air.

Some time ago when air conditioning was first installed in office suites, it was the consensus that grilles should be provided in all inner office doors in order to allow for the exhaust air. I have found that most inner office doors are equipped with transoms which are oftentimes open, or that by far the largest per cent of the time, the doors of these offices are left open, and this allows for the movement of air. I have found that where grilles have been installed in the doors, the occupant of that office often wants absolute privacy when talking to a client and it is very hard to have absolute privacy with a grille in the door panel. Considering the fact that very few meetings of this type are of very long duration, it does no harm to leave the door closed.

Another important point to be remembered when conditioning an office of this type is that of air velocity. The cost of sheet metal and grilles in proportion to the whole installation is small and some otherwise very good jobs have been spoiled by cutting down the size of the duct work and the grilles with the result of air delivery at high velocity which causes great discomfort to the occupants of the room. I have found that a grille installed in a private office, directly opposite a window, and discharging at a velocity that will carry this air to the window, will cause great discomfort to the occupant sitting at his desk during the winter, as when the air hits the cold glass, it seems to take a running jump for the fellow's neck who happens to be sitting anywhere near it, and most desks are backed up fairly close to the windows.

Every man in the air conditioning business has his own ideas regarding the installation of his equipment and he sincerely believes them to be right. The above facts were learned from experience after "living with" and operating various types of systems for some time.—Building Manager.

[The editors wish to express their appreciation to those readers who submitted replies to this question. Others which do not appear here were of considerable merit and would deserve publication if more space were available.]

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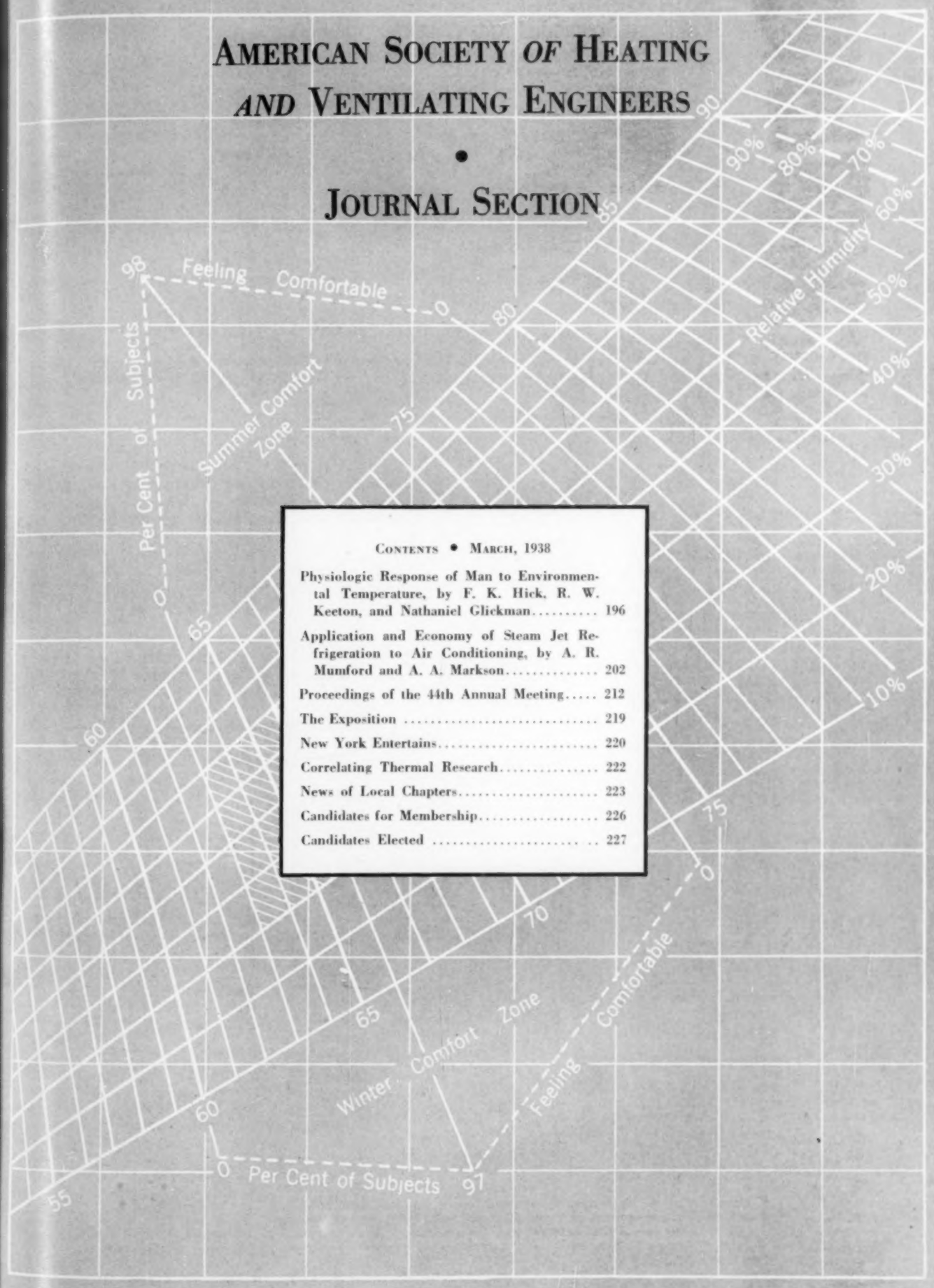

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Physiologic Response of Man to Environmental Temperature

By F. K. Hick, M. D., Ph.D.,* R. W. Keeton, M.D.** and Nathaniel Glickman, M. S.,*** *(NON-MEMBERS)*

Chicago, Ill.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the University of Illinois, College of Medicine

THE response of the body to atmospheric environment has long been a subject of interest, which has filled literature with important observations. In the earlier studies investigators have taken advantage of fortuitous circumstances, which constituted conditions for a controlled experiment. Thus Hunt,[1] from studies in hot dry climates, made observations on the quantity of water which must be evaporated to render the body comfortable. Haldane,[2] working under conditions of high humidity, fixed the temperature (88 F expressed as a wet-bulb), at which the dissipation of heat was inadequate and the body developed a fever. Here also should be noted the numerous observations upon heat stroke.[3, 4] It is obvious that the outlook for progress from studies of this type was limited, since chance only rarely sets up conditions for a controlled experiment.

Another approach to this subject (Petersen,[5] Mills[6 a,b,c,d]) is found in the study of the effect of climate and weather on the individual. Since it is often difficult to measure quantitatively the adjustments of normal individuals to these changes, patients with disease were selected. The course of a given disease, with its periods of easily discernible invasion and remission, has been charted. The weather records covering the same period have been studied and the variable factors traced. Where a given weather variable has shown a close correlation with the swings of disease, it has been considered significant in modifying the body's response. This important type of observation has already served to focus attention on factors which must be subjected to experimental study.

For an experimental approach to the problem, the capacity of the environment to transmit and dissipate heat should be known or measurable. Hill[7] introduced the Kata thermometer for this purpose, and it was pointed out by Winslow[8] that changes in this instrument are an approximate index of bodily comfort. Somewhat later other studies (Houghten[9]) led to the development of the *effective temperature* scale, which is now in general use. This scale integrates the effects of temperature, humidity and air velocity. Winslow and associates[10] have now arrived at an expression for the radiation properties of the environment. It would seem therefore that in the near future some agreement will be reached as to the definition and analysis of the heat dissipating capacity of the environment. Until such is obtained, workers will continue to speak different languages and interpret their results diversely.

When one turns to the behavior of the human body, factors are found which render the problem more complex. The oxidative processes develop heat, and the body dissipates it, thus modifying the environmental heat capacity. This constitutes an effect of the body on its environment. The body, in turn, must lose a given amount of heat, or the heat will accumulate and elevate the body temperature, thus producing a fever. The machinery used for this purpose is complex. The methods employed vary widely, but they are conditioned by the capacity of the atmosphere for heat dissipation. Under one set of conditions the body's heat may be lost entirely by evaporation and under other conditions chiefly by radiation. In these adjustments an effect of the environment on the body is produced.

The problem of an individual's adjustment to environment is essentially one in physiology, and must be approached from this experimental viewpoint. DuBois[11] and Winslow et al.[12 a,b] have recently analyzed quantitatively the forms of heat loss of the body for a rather wide range of temperatures and humidity. The large role played by circulation in making it possible for the body to lose heat under a variety of circumstances has become apparent to all workers. Under one set of conditions the skin and periphery is warm and moist, and under another dry and cool. At times the pulse rate is speeded up, and the individual complains of over-

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Fig. 1—Skin temperature measurements on subject in air conditioned room

action of the heart. On a hot day people are driven to drink increased quantities of water by thirst, a physiologic device designed to replenish blood volume. These and similar observations are familiar to all. An analysis of such adjustments must be made. This includes changes in heart action, blood vessel behavior, blood volume, and blood distribution. In the present study are reported certain objective measurements, which are to be regarded as preliminary to a major objective, the adjustment of the circulation to the environment.

Experimental Details

Subjects. This study was performed with the assistance of the house officer staff at the Illinois Research and Educational Hospital located in Chicago. Since normal subjects are scarcely available for hospitalization for weeks at a time, patients were used having certain complaints which required hospitalization and yet had no abnormality in heat regulation which could be detected. It is felt that the responses of these subjects are essentially those of normal subjects. The subject I.W. was a healthy male of 24, a medical student. The other subjects had been hospitalized for some time, were ambulatory, and were psychologically suited to the restriction of activity required of them. They were sufficiently trained in the procedures used to make their reponses reliable. The subjects are designated by initial, sex mark, age and diagnosis on the illustrations.

Observations. The variables on which daily observations were made were: basal metabolism or heat production, vital capacity, skin temperature and rectal temperature, and the presence of sweating or shivering.

Procedure. The air conditioned room is a small chamber accommodating two beds on the medicine floor of the Illinois Research and Educational Hospital. The conditions within the room were checked daily at several places and different times by a sling psychrometer. Air velocity was measured by a vane type anemometer recording an air movement of about 55 fpm. The fans were operated at a constant speed.

Two subjects dressed in standard cotton pajamas entered the room in the evening and made themselves comfortable under suitable bed clothing. About 6 a.m. they were awakened, the covers thrown back and they lay quietly for 1 to 2 hours before measurements for that day were begun (Fig. 1). In order to standardize costume the subjects rolled up the sleeves and legs of the pajamas as shown and had the trunk largely exposed.

First, two measures were made of

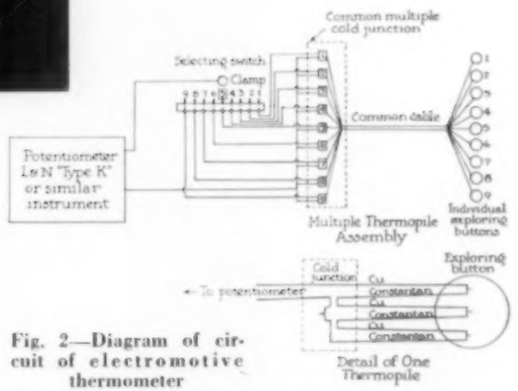

Fig. 2—Diagram of circuit of electromotive thermometer

the heat produced per unit time, basal metabolism. Then skin temperature measurement was carried out, and the rectal temperature taken. Finally the vital capacity was measured.

Skin temperature measurements were made by means of a thermopile arrangement having several exploring or unknown temperature junctions. The electrical circuit is diagrammed in Fig. 2. The cold junctions, each consisting of 3 copper constantan junctions, were kept beside a mercury thermometer. Copper wire was wrapped about them all, then they were introduced into a test tube which was kept in a thermos jar filled with ice and water. The exploring button was patterned after the apparatus of Sheard.[13] The circuit differed from Sheard's in having a different switch and in the use of a potentiometer as a null point instrument rather than using a deflection galvanometer. The exploring buttons were held on the skin by narrow strips of adhesive. This apparatus carries the usual probable errors of similar devices. It depends on contact with the skin, yet may by contact insulate the skin and so alter the temperature and/or alter the capillary circulation in the skin under the button.

Daily readings of the skin temperature were made with the buttons on the same points: forehead, sternum, balls

[13]The Electromotive Thermometer: An Instrument and a Method for Measuring Intravenous, Superficial and Cavity Temperatures, by C. Sheard (*American Journal Clinical Pathology*, 1:209, 1931).

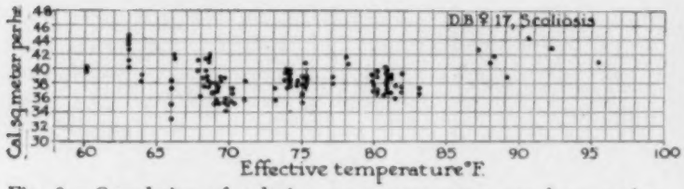

Fig. 3a—Correlation of calories per square meter per hour against effective temperature

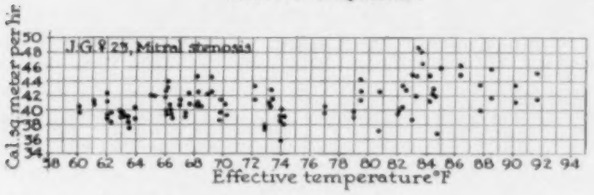

Fig. 3c—Correlation of calories per square meter per hour against effective temperature

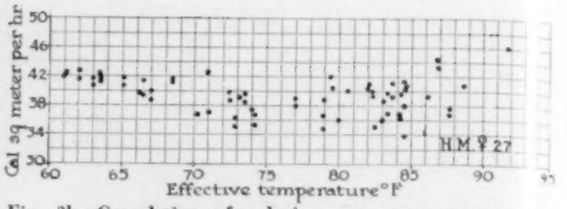

Fig. 3b—Correlation of calories per square meter per hour against effective temperature

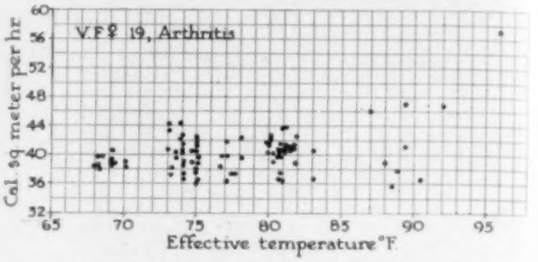

Fig. 3d—Correlation of calories per square meter per hour against effective temperature

of both middle fingers, the right forearm, balls of both great toes, and a point above the right patella.

Rectal temperature was taken by an ordinary clinical thermometer.

Basal metabolism, the heat produced per unit time expressed as calories, per square meter per hour, requires that the subject be rested by a night's sleep, have had no food for over 12 hours and be relaxed. The Tissot tank method was used. In this procedure the expired air is collected, its volume measured, and its content of CO_2 and O_2 determined by analysis. This analysis permits the calculation of the respiratory quotient (R.Q.). The R.Q. is $\frac{CO_2 \text{ expired}}{O_2 \text{ inspired}}$. If the R.Q. is at wide variance from the normal, the situation is not *basal* and the results meaningless. A mouth piece and nose clip were regularly used.

The vital capacity is defined as the maximum volume of air a subject can exhale after the deepest breath he can take. A simple spirometer measures it. Although very different in the different subjects, it is relatively constant for each individual. Although a few other factors control vital capacity, it largely mirrors the amount of blood in the lungs. The greater the volume of blood in the lungs, the less air they can hold.

Sweating and shivering were sought for by inspection and recorded daily.

Results

Heat Production. Basal metabolism is recorded on the charts in Fig. 3, where the effective temperature is the abscissa and the heat production expressed in calories per square meter per hour is the ordinate. (A calorie per square meter per hour is equivalent to 0.369 Btu per square foot per hour.) In such distribution charts there is a scatter of points on either side of a trend line. If the heat production were a function of the effective temperature, some correlation should be apparent at a glance. If perfect correlation were obtained, all the points would lie on a single line. The points are least widespread when effective temperature is the abscissa. When the dry-bulb (Fig. 4) is used on the abscissa, the scatter is a bit wider, though not greatly different. Similarly, the basal metabolism is not a function of the wet-bulb.

Fig. 3a shows an apparent slight rise in the heat production at the cool end of the curve. In this subject shivering was noted when effective temperatures below 65 deg were obtained. However, the rise in the curve does not prove significant on statistical analysis of this (and all the other) curves. The variations observed in the basals were too great to permit this slight rise to bear significance.

Fig. 3b illustrates also the slight rise in heat production with shivering in a cold atmosphere.

Fig. 3c, J.G., a woman of 23 who had survived severe heart failure, did not show any rise in metabolism in the cooler ranges. She did not shiver. It seems that her basal metabolism and body temperature began to rise at environmental temperatures lower than that of normal subjects.

Fig. 3d does not extend low enough to involve shivering.

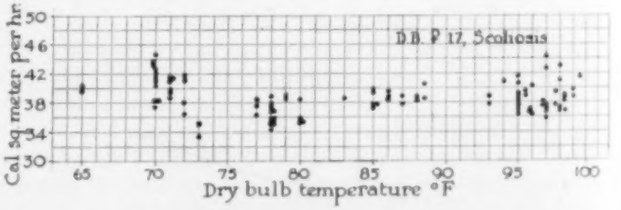

Fig. 4—Correlation of calories per square meter per hour against dry-bulb

All of these measurements were determined with the subjects at *basal* conditions. Under these conditions the heat produced becomes minimal for that individual.

Rubner[14] measured with a calorimeter the heat production of a dog kept at different dry-bulb temperatures. In the cold zone, below 16 C the animal's heat production rose to values greatly in excess of those observed at temperatures from 20 to 30 C. Fever was observed with temperatures above 35 C in the environment. The *chemical regulation* of body temperature was distinguished from the physical regulation.

There has been some dispute as to the existence of a chemical regulation of heat in man. Loewy,[15] among

[14]Die Beziehungen der atmospharischen Feuchtigkeit zur Wasserdampfabgabe, by M. Rubner (*Arch. f. Hyg.*, 11:208, 1890).

[15]Ueber den Einfluss der Abkuhlung auf den Gaswechsel des Menschen, by A. Loewy (*Pfluger's Arch.*, 46:189, 1890).

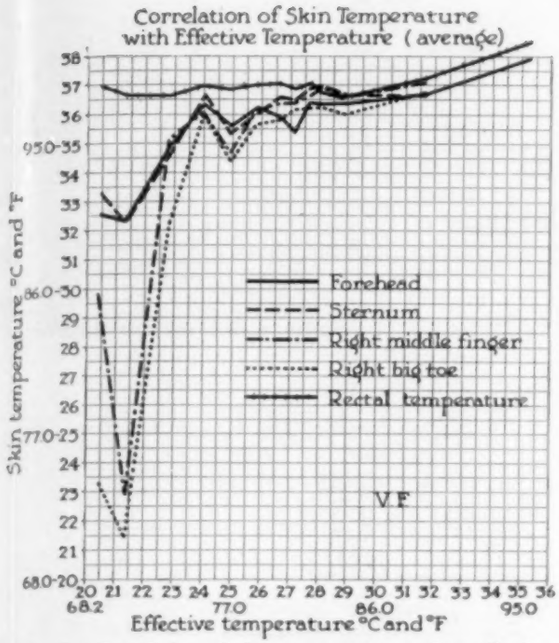

Fig. 5a—Correlation of skin temperature against effective temperature

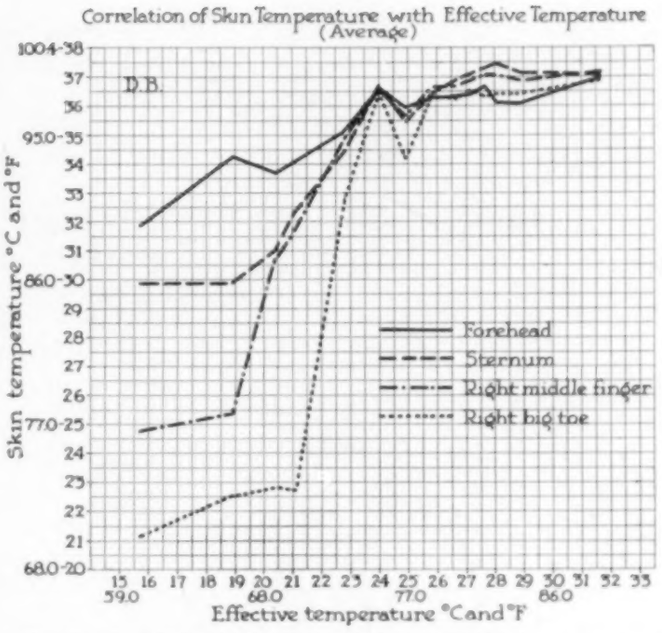

Fig. 5b—Correlation of skin temperature against effective temperature

others (Johansson[16]) concluded that shivering and other movements might lead to increased heat production. In the absence of shivering, the heat production was unchanged. Hill[17a,b] came to the opposite conclusion.

McConnell and Yaglou[18] did basal metabolic rate determinations on healthy men, and a curve was developed plotting basal metabolism against effective temperature as the abscissa. This curve rises at either end, and the rise diagrammed in the hot range is similarly suggested in the data reported in this paper. Winslow et al.[19] comment that the metabolism remains constant throughout the range of experimental temperatures observed. The subjects had food 2 hours before the determination was started. Digestion is associated with a variable rise in heat production of over 10 per cent.

The rise in either end of the temperature ranges studied by McConnell and Yaglou may partly result from the inability of subjects to remain quiet and relaxed when distressingly chilled, or acquiring a fever. In the one instance shivering is involuntary; in the other the subjects are increasingly restless. Experience acquired in this study supports this opinion. It is established that when fever heat appears, heat production rises in accord with van't Hoff's law.

The literature does not mention any man capable of lowering his heat production as the environment becomes too warm under acute conditions.

Data accumulated by animal experiments cannot be transferred to man when they deal with heat regulation. Mammals as a rule are insulated with hair, cannot sweat, or do so in small degree. They cannot readily vary their heat loss by radiation as man does. Therefore, mammals require some chemical regulation of heat production which man has replaced by better means of heat loss (sweating and variable radiation).

Data obtained in these observations support the contention that chemical regulation of body temperature in man plays an insignificant role within the ranges shown.

Results of Skin Temperature Measurement

Skin temperature results are illustrated by the graphs in Fig. 5, where the abscissa is effective temperature. It is apparent that in the cooler zone the extremities, particularly the hands and feet, cool down, even being observed lower than room temperature. The temperature of the hand rises when the environment is somewhat warmer. When still warmer environment is present, the feet rise in temperature, reaching a peak about blood temperature, with an effective temperature of 75 deg. The temperatures of the sternum and forehead, upper arms and thighs diminish as the temperature of the environment falls. They do not show variations comparable in extent with the hands and feet. Above 82 deg ET the extremities continue to rise slowly, even exceeding finally the sternum and forehead.

At 77 deg ET there is observed a transient fall in temperature of the extremities, especially the hands and feet. Since this is slightly below the point at which sweat appears grossly on the skin, it is considered to be a change in the radiation of heat, while imperceptible perspiration comes out on the skin and assists with body cooling.

The work of Maddock and Coller[20a,b] and Sheard[21] established the variable radiation capacities of the ex-

[16]Ueber die Tagesschwankungen des Sloffwechsels und der Korpertemperatur in nüchtenem Zustande und vollständiger Muskelruhe, by J. E. Johansson (*Skandin. Arch. für Physiol.*, 7:123, 1897).

[17a]. Report on Ventilation and the Effect of Open Air on the Respiratory Metabolism, by L. Hill (Local Gov. Bd. Report, New Ser., No. 100, 1914).

[17b]. Observations on the Resting Metabolism of Children and Adults in Switzerland, by L. Hill and J. A. Campbell (*British Medical Journal*, 1:385, 1922).

[18]Basal Metabolism as Affected by Atmospheric Conditions, by W. J. McConnell and C. P. Yaglooglou (*Archives of Internal Medicine*, 36:382, 1925).

[19]Loc. Cit. See Note 10.

[20a]. Role of the Extremities in the Dissipation of Heat, by W. G. Maddock and F. A. Coller (*American Journal Physiology*, 106:589, 1933).

[20b]. The Function of Peripheral Vasoconstriction, by F. A. Coller and W. G. Maddock (*Annals of Surgery*, 100:983, 1934).

[21]Effect of Changes in Environmental Conditions on Skin Temperatures and the Dissipation of Heat from the Body, by C. Sheard, M. D. Williams and B. T. Horton (*Proceedings American Physiological Society*, p. 147, 1937).

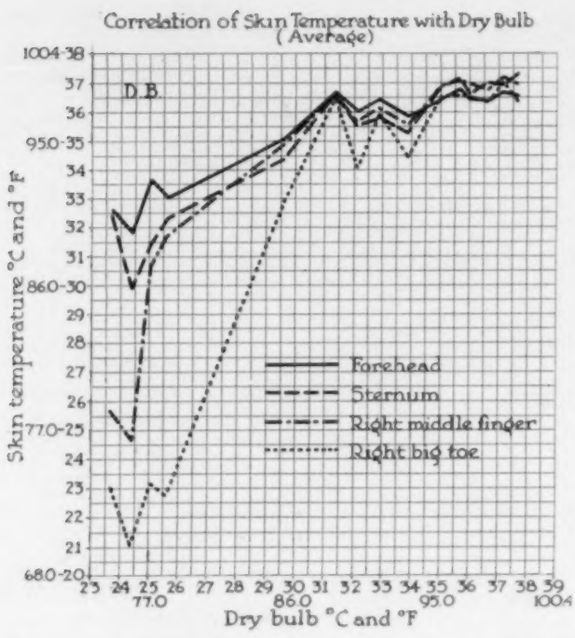

Fig. 6—Correlation of skin temperature against dry-bulb

tremities. Barbour[22] demonstrated the reflex regulation of skin temperature. Hardy[23] has measured the great sensitivity of the skin to application of radiant heat.

It is clear that a mechanism exists which regulates skin temperature so as to control heat loss by radiation. In the zone 66.0 to 72.5 deg ET this control rests largely in the hands. In the higher range, 70.0 to 75.0 deg ET, it is in the feet.

That these data can be best correlated with the effective temperature is not surprising. Fig. 6 indicates the less regular distribution of points when skin temperature is plotted against dry-bulb. It is well known that the amount of heat lost from the body by radiation depends on the temperature and character of the walls; that the heat lost by evaporation depends on the wet-bulb, the dry-bulb, air velocity and type of air movement; and the heat lost by conduction and convection on dry-bulb and air motion. It was anticipated that the effective temperature in combining three of these variables would come nearer correlation with physiologic observations.

Sweating

The onset of sweating in these subjects (Fig. 7) was about 81 deg ET. This was rather constant where no disease process was significant in control of heat loss. The fact that these subjects were completely at rest and fasting permitted making this observation of the critical temperature for appearance of sweating. If the body's heat production should increase in response to exercise or the specific dynamic action of food, the onset of sweating would be at lower temperatures.

[22]The Heat-Regulating Mechanism of the Body, by H. G. Barbour (*Physiological Review*, 1:295, 1921).

[23]Studies in Temperature Sensation III. The Sensitivity of the Body to Heat and the Spatial Summation of End Organ Responses, by J. D. Hardy and T. W. Oppel (*Journal of Clinical Investigation*, 16:533, 1937).

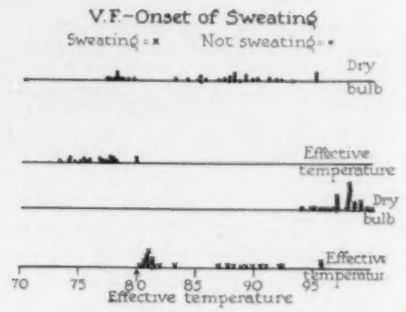

Fig. 7a—Correlation of sweating against effective temperature

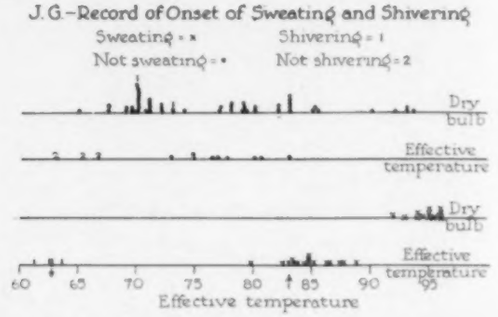

Fig. 7b—Correlation of sweating against effective temperature

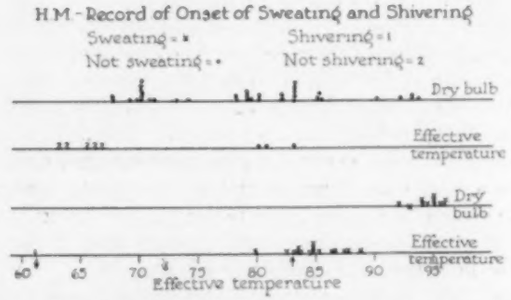

Fig. 7c—Correlation of sweating against effective temperature

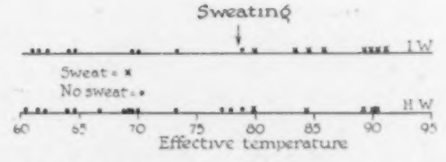

Fig. 7d—Correlation of sweating against effective temperature

Shivering

The onset of shivering was not indicated at any critical temperature. In general it was observed below 65 deg ET

Vital Capacity

The results of vital capacity measurements are shown with scatter charts using effective temperature as the abscissa (Fig. 8). There is a consistent and significant

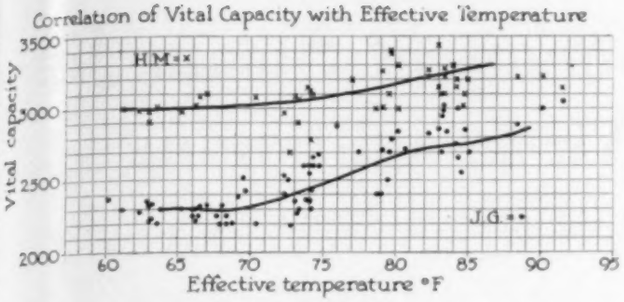

Fig. 8a—Correlation of vital capacity against effective temperature

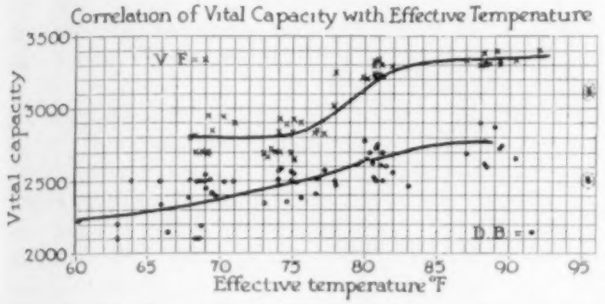

Fig. 8b—Correlation of vital capacity against effective temperature

rise in vital capacity as the temperature rises. The rise exceeded 1000 cc in I. W. a healthy medical student and 800 cc for J. G. a girl with organic heart disease. The subjects were not aware of any change in sensations as their vital capacity rose. On the days when conditions were between 70 and 80 deg ET they expressed the opinion that their vital capacity would be higher. This prediction was regularly in error.

It is the authors' opinion that this rise in vital capacity reflects a shift in the distribution of blood within the body. An increased volume of blood in the skin would lead to a reduced quantity somewhere else. Blood reservoirs in the lungs[24 a, b, c] and abdomen could supply this blood. The work of Moore and Kinsman[25] included measurement of the volume of blood within the chest and observed results which are supportive of the idea that the lungs normally act as a reservoir for storage of blood.

Hyperventilation

Hyperventilation was noted as a response of some subjects to hot environments. Trained subjects, exposed to very high ET, breathe out per unit time a great deal more air than at lower ranges. This response is often given by nervous, and untrained individuals at lower temperatures. It is a common source of error in observations on untrained subjects against which suitable precautions must be taken. It was observed in these tests when the body temperature was rising.

When hyperventilation lasts over 2 min there is enough CO_2 washed out of the body to give rise to symptoms of headache, nausea, weakness, giddiness and prostration. In fact hyperventilation tetany is described with numbness and tingling in the extremities followed by characteristic muscle spasm.[26 a, b] This may be one of the mechanisms of producing discomfort in high environmental temperatures.

Conclusions

1. The *chemical* regulation of body temperature is negligible in man under basal conditions and within the ranges investigated.
2. Radiation of heat is a principal means of heat loss in a range below 80 deg ET. It is controlled by a sensitive regulation of skin temperature in the extremities.
3. The point at which sweating appears for a subject under basal conditions is about 81 deg ET. A drop in skin temperature of the extremities occurring a few degrees below that suggests that evaporation from the body surface has increased at 77 deg ET.
4. Under basal conditions rectal temperature does not rise until the effective temperature reaches 91 deg.
5. Evidence is presented of a redistribution of blood as the skin temperature rises, leading to a diminution of blood content of the lungs.
6. Hyperventilation is a transient response to intolerable heat.
7. The effective temperature scale comes nearest to correlation with the observations reported in these investigations. If to this could be added the factor of radiation capacity of the environment, better correlation could be expected.

[24a] The Effect of Pulmonary Congestion on the Ventilation of the Lungs, by C. K. Drinker, F. W. Peabody and H. L. Blumgart (*Journal Experimental Medicine*, 35:77, 1922).

[24b] Failure of the Circulation, by T. R. Harrison (Williams and Wilkins, Baltimore, p. 131, 1935).

[24c] Architectonics of Vessels of Lungs with Special Consideration of Blood Storing Function of Lungs, by R. A. Pfeifer (*Ztsch. f. Kreislaufforsch*, 26:906, 1934).

[25] Studies on the Circulation: Further Analysis of Injection Method and of Changes in Hemodynamics under Physiological and Pathological Conditions, by W. F. Hamilton, J. W. Moore, M. M. Kinsman and R. G. Spurling (*American Journal Physiology*, 99:534, 1932).

[26a] A Study of Forced Respiration. Experimental Production of Tetany, by S. B. Grant, and A. Goldman (*American Journal Physiology*, 52:209, 1920).

[26b] The Effect of Prolonged Hyperpnea on the CO_2 Combining Power of the Plasma, the CO_2 content of Alveolar air and the Excretion of acid and basic phosphate and ammonia by the kidney, by J. B. Collip and P. L. Bachus (*American Journal Physiology*, 51:658, 1920).

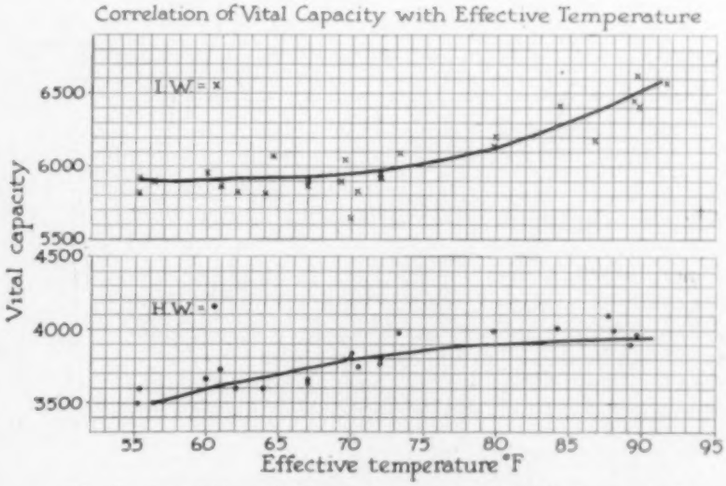

Fig. 8c—Correlation of vital capacity against effective temperature

Application and Economy of Steam Jet Refrigeration to Air Conditioning

By A. R. Mumford* and A. A. Markson**, (*NON-MEMBERS*) New York, N. Y.

WITH the introduction of steam jet refrigeration, district heating companies became interested in it as a potential source of summer load and improvement in annual load factor.

On Fig. 1 are plotted the monthly total sales of steam to consumers of the New York Steam Corp. for the calendar year 1936. Two dotted sections have been added to the curve to indicate the increase in summer load if 10 per cent or 20 per cent of the space now served with heating steam should also be cooled by steam jet refrigeration. The ratio of the average month to the peak month is shown below each curve as a percentage. Although this is not the usual load factor, it is a relative figure and indicates a possible improvement which ought to be mutually beneficial. As in every new development, the particular conditions for which steam jet had to be designed were not fully understood in the beginning. The first machines were sold on the basis of

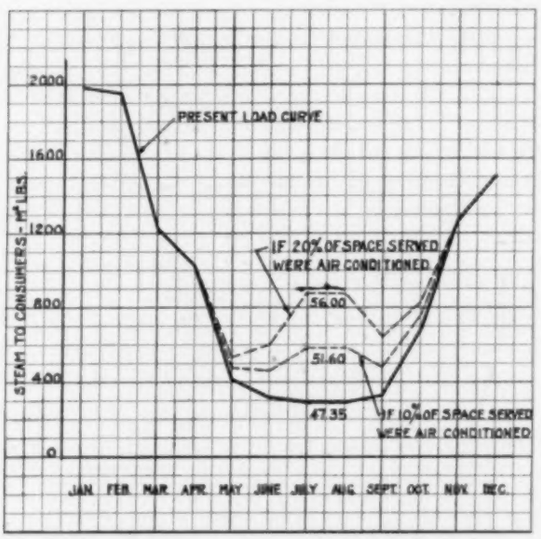

Fig. 1—Estimated effect of summer load which might be added by steam jet refrigeration

first cost and simplicity. The tendency was to model the installations after the well established features of compression machines. There are 18 machines now operating on the utility service in New York City, aggregating 1560 tons in capacity. Most of these were satisfactory as installed, but in some cases the utility had to work on some of them with the cooperation of the manufacturers. The results of this effort have been to help establish this type of refrigeration on a sound basis. In order to be able to assist those of its consumers who had installed such equipment or who were interested in it, a flexible program of tests and investigations was introduced to provide the basic information for sales and service in cooperation with the manufacturers and consulting engineers.

As compared with other refrigerants being used for air cooling, water vapor may be said to come into its own, thermodynamically, above temperatures of 40 F. Air cooling consists of removing heat from an air-water vapor mixture. This heat, in the form of latent and sensible, is then pumped by a heat pump to the final heat level. In doing this practically, it is important to realize that the heat may have been transformed several times from one form to another merely for the convenience of the particular method. The steam jet machine as applied to air cooling may be considered as one in which a certain amount of space heat, sensible and latent, is first converted into the same quantity of heat entirely sensible, which heat is then pumped to the condenser in the form of water vapor or latent heat.

Steam jet cooling may therefore be regarded as a process of air cooling in which refrigeration is done on water vapor and air by means of water vapor and air without the use of special refrigerants to act as heat carriers. Below certain cooling temperatures a refrigerant of better thermodynamic properties than water vapor may be required. Thus, one of the main points of this paper is the discussion and demonstration of the air conditioning applications which fall into the economical range of water vapor refrigeration. It is believed that consideration of these demonstrations will inform engineers familiar with steam jet that objections to steam jet, based on the specifications of very low water temperatures, arise only when its operation is improperly compared with that of older systems.

Secondly, it is the purpose of this paper to demonstrate that steam jet as a direct process should be treated as such in its application in order to obtain real operating advantages and economies.

Third, a thermodynamic analysis is presented of the jet itself to familiarize engineers with the load characteristics of these machines obtainable by reduced steam pressure operation at partial cooling tower loads and the controls by which these are obtained.

Last, a discussion is given of the really fundamental practical matter of condenser water from a constructive engineering viewpoint.

Selection of Dew-Point or Flash Tank Temperature

The selection of the proper refrigeration temperature depends on so many conditions that a discussion of them

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Presented at the 44th Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., January, 1938.

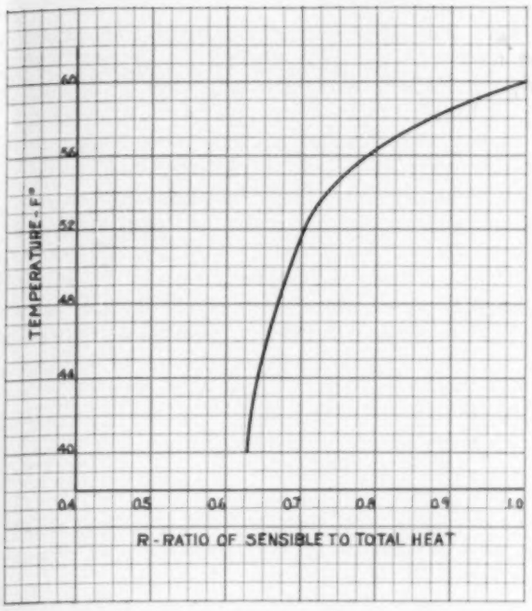

Fig. 2—Variation of dew-point with ratio of sensible to total heat picked up

all is beyond the scope of this paper. However, the factors which fix the proper dew-point can be discussed for two conditions, namely with and without the use of sensible reheat.

Without reheat, the dew-point is fixed by the ratio of sensible heat picked up in the conditioned space to total heat pickup. On Fig. 2 is shown a curve of dew-points which correspond to the heat ratios picked up in the conditioned space when the space is to be maintained at 80 F dry-bulb and 50 per cent relative humidity. For ratios much below 0.63 sensible to total heat pickup sensible reheat should be used. The limiting dew-point as shown is 60 F which is the highest allowable for 100 per cent sensible heat pickup under the maintained conditions of 80 F dry-bulb and 50 per cent relative humidity. As more and more latent heat is picked up the maximum permissible dew-point drops until a practical minimum of 40 F is reached at the 63 per cent ratio.

If the ratio of sensible heat to total pickup in a conditioned space is considered as the determinant of the character of the load to distinguish it from the size of the load, it can be stated that the character of the load determines the only possible dew-point which can be used without reheat to meet any fixed condition to be maintained in the space. To illustrate the relation of dew-point to load characteristic, a series of psychrometric charts[1] are presented which are peculiarly well suited for the purpose.

If on a standard psychrometric chart plotted with moisture content per pound of dry air against dry-bulb temperature, a line be drawn between a point in the diagram and a point on the saturation line, it is seen that the ratio of latent to sensible heat is practically constant along this line. The chart illustrated in this paper is a specialization of this principle in which the psychrometric chart has been erected on a framework of such ratio lines (sensible to total heat) yielding a clear method of analyzing practical psychrometric problems. As applied to these problems, this chart has an appeal similar to that of the Mollier steam diagram in steam power problems.

On Fig. 3 is indicated a fixed condition (Point 1) in the comfort zone at 80 F dry-bulb and 50 per cent relative humidity which it is desired to maintain within the conditioned space. The washer dew-point (Point 2) was selected arbitrarily at 40 F and the line connecting the fixed condition with the dew-point is the locus of all points of mixtures of air, recirculated and conditioned, which will satisfy the desired conditions without reheat. On this type of chart, if a line is drawn through the reference center intersecting the arc of ratios at the upper part of the chart, and parallel to the line 1—2, the point of intersection is the unique ratio of sensible to total heat pickup in the conditioned space which will permit the desired condition to be maintained with a 40 F dew-point and without reheat. If it is assumed, for instance, that a new reference center line, *A*, is drawn, showing an increase in sensible heat pickup in the conditioned space, it is obvious that its parallel, *A'*, through the desired condition will intersect the saturation line at a higher temperature and will require a higher dew-point to satisfy the conditions. Similarly, if line *B* is drawn through the reference center corresponding to an increase in latent heat pickup in the conditioned space, its parallel, *B'*, will intersect the saturation line at some dew-point temperature below 40 F or not at all. Thus, it can be stated that for a fixed condition to be maintained with a fixed dew-point only one character of load may be met. Conversely, for a fixed condition, the dew-point must be varied to correspond with changes in the character of the load.

To examine this relation of dew-point to load character further, the original assumptions were changed to permit variation in the dew-point. On Fig. 4 is shown the same fixed condition to be maintained in the conditioned space without reheat. The lines *A* and *B* from the fixed condition to the saturation line represent the limiting lines of mixtures. The line *A* is drawn to the practical lowest dew-point of 40 F and the line *B* to the highest possible dew-point at about 60 F. The reference center parallels, *A'* and *B'*, indicate that the character of the load may change from a ratio of 100 per cent sensible to total heat pickup to one of about 63 per cent. Within this range of load character, it is possible, therefore, to maintain the desired conditions in the space exactly by raising the dew-point as the proportion of sensible heat pickup increases.

Not only is the advantage of maintaining exact conditions obtained but it is possible to operate at increasing economy as the dew-point rises because of the flash tank temperature—steam consumption characteristic of the steam jet machine.

If assumptions of fixed conditions are replaced with reasonable allowances for variations, possible means are developed for improving operating economy. The assumptions for Fig. 5 are that conditions be maintained in the space at any point along the effective temperature line, that no reheat is to be used, but that any suitable dew-point may be selected for the operation under consideration. Conditions have been assumed to be maintained on an effective temperature line rather than that

[1]The Psychrograph, by A. M. Norris (*A.S.M.E. Transactions*, July, 1935).

they be allowed to stray throughout the comtort zone under the impression that wide variations, though within the comfort zone, may not result in complete satisfaction. Dew-points of 40 and 55 F have been selected as representing low and high dew-point operation. Limiting lines of mixtures have been drawn to the dew-points from the ends of the effective temperature line. The corresponding parallels have been drawn from the reference center to the ratio arc and these intersections indicate the ranges of load character which may occur in the conditioned space and yet have the maintained condition remain somewhere along the effective temperature line. Both the position of the range and the size of the range are different for the two dew-points. The higher dew-point, as would be expected, can not handle as great a latent heat load without reheat as the lower dew-point. Conversely, the lower dew-point can not handle as high a proportion of sensible heat as the higher dew-point.

The approximate changes in the character of the load which may take place in theaters, restaurants, office buildings and stores are shown in Table 1.

Table 1—Estimate of Range of Change of Load Character Which Installations Should Be Expected to Handle

Type of Occupancy	Ratio of Sensible Heat to Total Heat Picked Up
Ballroom, Night Club, etc.	54—80
Theater Building	70—90
Store	74—90
Office	78—90

One of the important variables is the person load which, in most cases, is the principal source of latent heat load. It is obvious that this load will vary during the day and, if comfort conditions are to be carefully maintained without reheat, the dew-point must be changed to suit the new ratio. It is to be noted that control along the effective temperature line within the comfort zone requires that the dry-bulb temperature vary from about 81.5 F at 40 per cent relative humidity to 78.5 F at 60 per cent relative humidity. The volume of air circulation is assumed to be constant so that variations of dry-bulb temperature and relative humidity result from changes in the proportion of recirculation and dew-point.

Fig. 6 indicates the ranges of load character which may be handled without reheat when the maintained conditions are allowed to vary along the effective temperature line and the dew-point is made variable from a practical low of 40 F to a maximum of 63.5 F corresponding to the 60 per cent relative humidity point on the effective temperature line. It is shown that by varying the dew-point, a change in load character of from 55 per cent sensible heat to 100 per cent sensible heat is possible in maintaining the desired conditions without reheat. By maintaining the maximum dew-point for any given load character, the greatest economy of refrigerating energy will be realized. The range of load character permitted by this scheme of operation is sufficient to meet those changes typical of most commercial applications.

On Fig. 7 reheat is introduced to the problem. It is assumed that a fixed condition of 80 F dry-bulb and 50 per cent relative humidity is maintained in the conditioned space with variable dew-point and reheat. At the lowest dew-point of 40 F, air by-passing is all that is required and a load character of 63 per cent sensible heat is the only one it is possible to meet. This ratio, however, can be met by a dew-point of 56 F by reheating from the saturation line to 70 F and eliminating air by-passing. For higher pickup ratios the treatment is the same as for the 63 per cent ratio and the maximum dew-point will rise with each ratio until, at 100 per cent sensible heat pickup, the dew-point is about 60 F. At any pickup ratio between 63 and 100 per cent the use of reheat permits the use of a higher minimum dew-point. Thus, where a set of conditions has been assumed which, without the use of sensible reheat, would call for a theoretical dew-point variation of from 40 to 60 F, it is apparent that the same conditions can be maintained by a dew-point variation of only 4 F, from 56 to 60 F, if controlled reheat is provided. It is apparent that large washers are required for these schemes of meeting load character differences but it is believed that the economy of high dew-point operation with steam jet equipment and the ability to maintain desired conditions exactly are factors which may well offset the increase in equipment size.

The range of load character shown in Fig. 7 can be extended by permitting the maintained conditions to follow along the line of effective temperature as shown in Fig. 8. It must be emphasized that sensible reheat from a comparatively high dew-point is required when the sensible heat constitutes less than about 63 per cent of the total heat picked up and the desired conditions can not be met by continuing to lower the dew-point.

No difference is made in the scheme of operation but the controls would, of course, be different. Some increase in economy of refrigerating energy is possible when the sensible heat ratio is high and the conditions are maintained at the lower end of the effective temperature line.

It has been indicated that the conditions required of all air cooling installations can be met by dew-points within the economical range of a steam jet refrigerator, most of them without reheat and some only if reheat is provided.

It has been indicated also that the attempt to meet the requirements of widely varying heat picked up ratios, such as are met in theaters, ballrooms and restaurants, without reheat and by the use of very low fixed dew-points is not theoretically correct and does not give good practical results. The low fixed dew-point arrangement is apparently an adaptation of mass production equipment to the problem.

It is believed that a simple and correct method of meeting variations in heat ratios picked up is afforded by direct dew-point control which is easily obtained with the steam jet machine.

If the dew-point, and therefore the refrigeration temperature, must be changed to meet exactly those changes in load character which occur in most air conditioning installations, it is important to know the influence of changing refrigeration temperatures on energy consumption. The influence of refrigeration temperature on the economy of steam jet is important so this characteristic is carefully examined in the following paragraphs.

The temperature to which the chilled water is cooled by flash evaporation is termed the flash tank temperature. In some machines, this temperature is held constant by thermostatic control. With such machines, which are designed to cool the circulating water between two fixed temperatures, it is necessary to vary the quantity of chilled water by-passed through the coils or

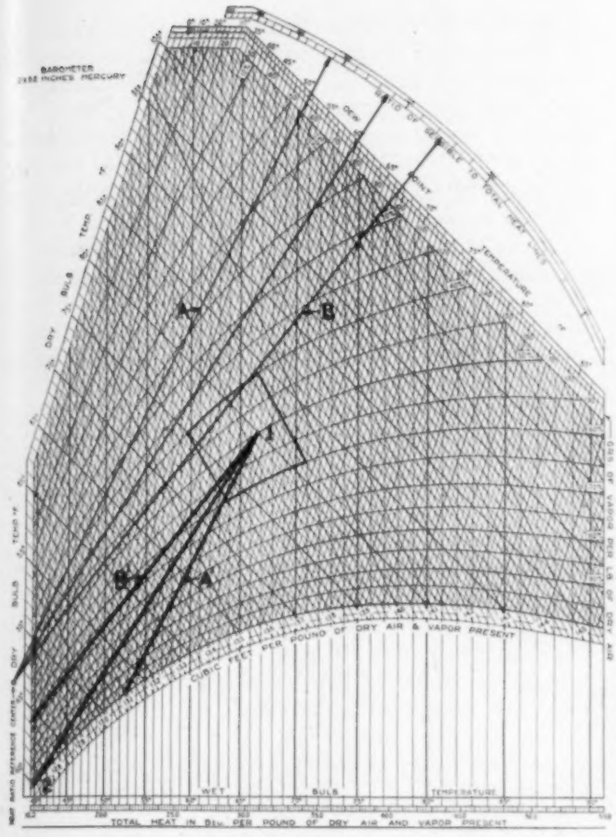

Fig. 3—A fixed condition with fixed pick up ratio can be met by only one dew-point without reheat

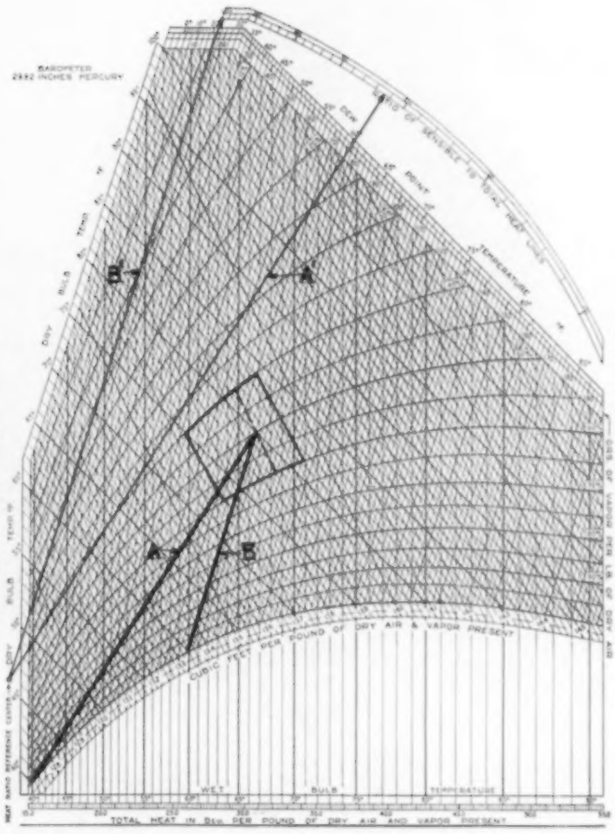

Fig. 4—Range of pick up ratios which can be met by varying the dew-point without reheat

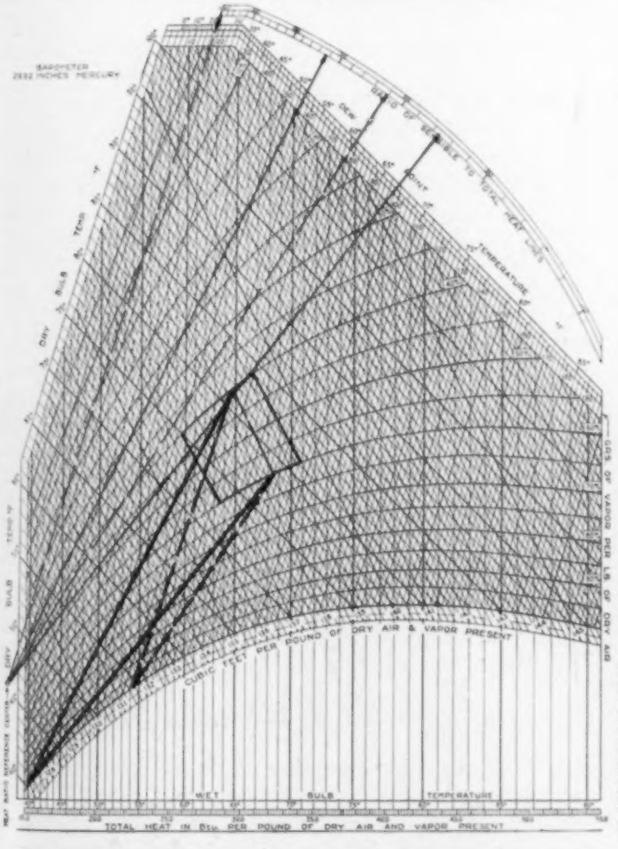

Fig. 5—Range of pick up ratios which can be met by low dew-point and by high dew-point without reheat if conditions are allowed to vary along effective temperature line

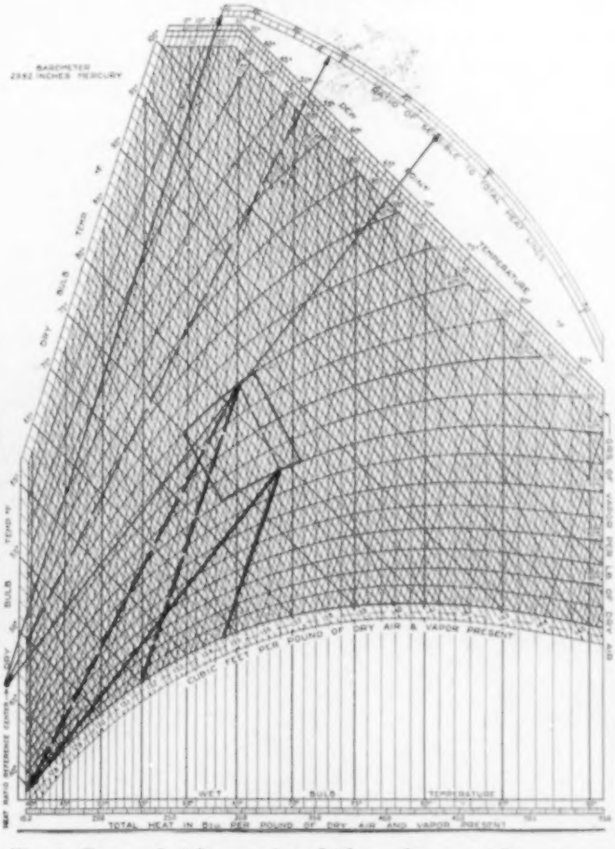

Fig. 6—Range of pick up ratios which can be met with maximum variation of dew-point without reheat if conditions are allowed to vary along effective temperature line

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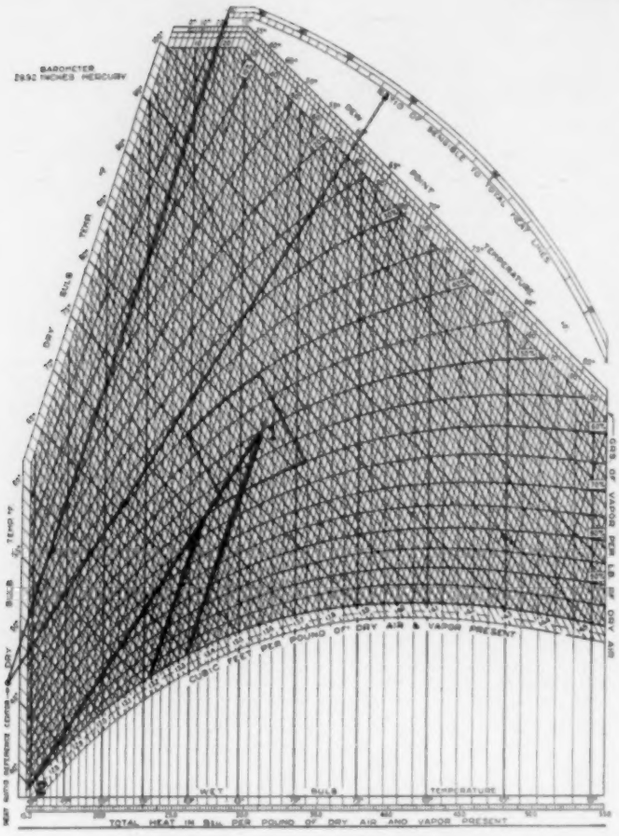

Fig. 7—Range of pick up ratios which can be met by varying the dew-point with reheat if conditions are fixed

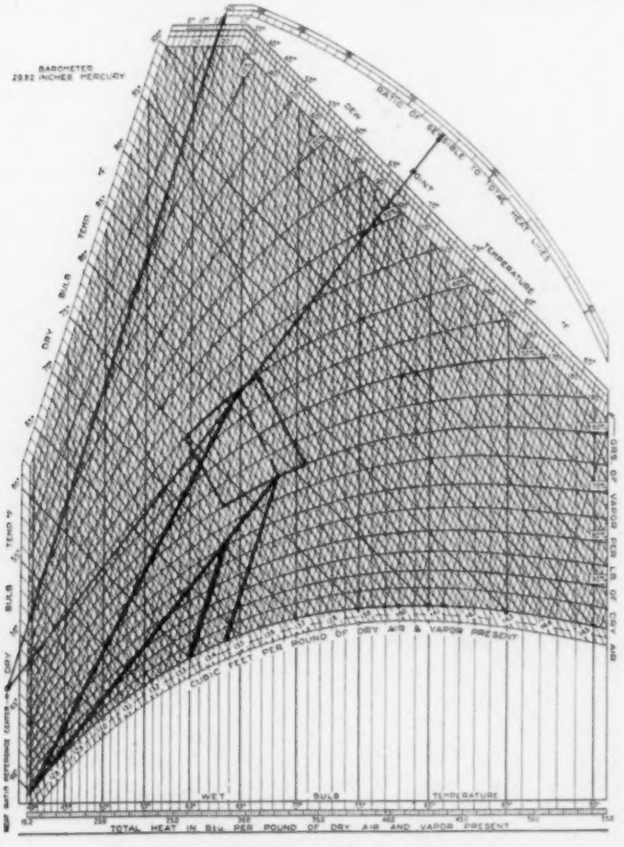

Fig. 8—Range of pick up ratios which can be met by varying the dew-point with reheat if conditions are allowed to vary along an effective temperature line

washer to meet variations in load. Other machines may maintain a fairly constant water temperature on the discharge side of the coils or washers by permitting the flash tank temperature to vary, depressing it as required by increases in washer load. With such machines, the quantity of chilled water circulated is held constant, the required variation in refrigerating capacity being obtained by the variation of temperature.

Theory, confirmed by tests of equipment installed on consumer's premises, indicated that the capacity of the steam jet machine to produce refrigeration increased with increase in flash tank temperature. For example, a booster capacity of 20 tons was obtained when the flash tank temperature was 40 F, and when it was 50 F a capacity of 32 tons was obtained. A little arithmetic will indicate readily why this result was obtained.

The booster system functions to remove water vapor from the flash tank, and to increase the pressure of this vapor to the pressure existing in the condenser where it is condensed, thus permitting the transfer of the heat content of the vapor to available condensing water. The heat of vaporization is withdrawn from the water in the flash tank and this removal results in a lowering of the liquid temperature in the tank. This withdrawal of heat sets up a differential between the heat in the liquid of the water entering the flash tank (H_1) and the water leaving the flash tank (H_2) which differential is made up of the product of the latent heat (L_2) and the fraction evaporated (X).

$$H_1 = H_2 + XL_2 \qquad (1)$$

The fraction evaporated (X) is obviously the quotient obtained by dividing the difference in heat contents by the latent heat per pound.

$$X = \frac{H_1 - H_2}{L} \qquad (2)$$

The useful heat rejected to or pumped up to the condenser is obviously $H_1 - H_2$.

The tonnage developed per pound of vapor is $\frac{L_2}{12{,}000}$ which corresponds to 11.3 lb of vapor per hour per ton of refrigeration at 50 F.

The specific volume of the vapor formed by the evaporation of part of the flash tank water decreases as the flash tank temperature increases, whereas the latent heat of evaporation remains nearly constant. Thus $\frac{L_2}{12{,}000}$ for 50 F is equivalent to 11.3 lb per ton as given previously or 19,200 cu ft of vapor must be handled per ton; whereas at 40 F, $\frac{L_2}{12{,}000}$ is equivalent to 11.2 lb per ton or to the handling of 27,400 cu ft of vapor per ton. Thus, on the basis of volumes alone, a booster system handling sufficient flash tank vapor to produce 20.0 tons of refrigeration at 40 F would handle enough to produce $\frac{27.4}{19.2} \times 20.0$ or 28.5 tons at 50 F. Actual tests bear out this conception of the jet as a constant volume compressor.

On Fig. 9 are shown the results of tests of refrigeration capacity as a function of flash tank temperature. The capacity of one booster is shown to rise almost as a straight line function of flash tank temperature. The

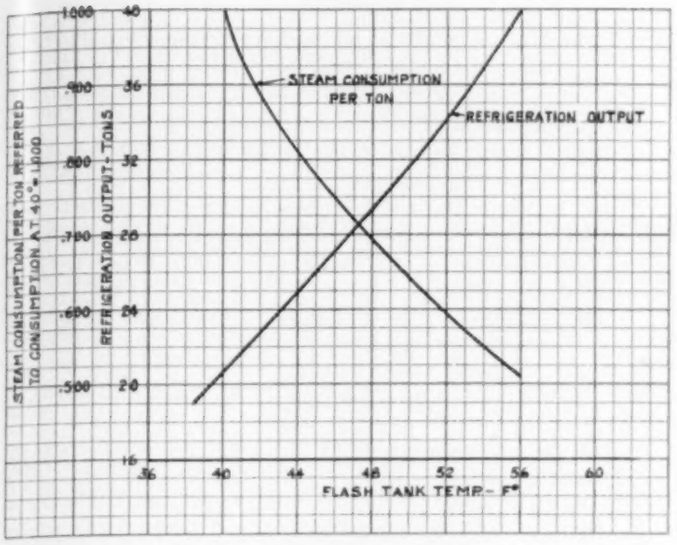

Fig. 9—Variation of steam consumption and refrigeration output with flash tank temperature

economy of high flash tank temperature operation is shown by the second curve which is almost a straight line function of flash tank temperature. The steam consumption has been expressed as a fraction of the consumption at 40 F flash tank temperature and it is to be noted that these fractions apply exactly only to the machine tested. However, except for variations in the design point, the fractions for other machines will follow the same characteristic curve. It is obvious that the rapid rate of increase of capacity or decrease in steam consumption with increase in flash tank temperature is a characteristic which should be borne in mind in the design of refrigeration for air conditioning.

The fact that the capacity of a steam jet machine increases with increased flash tank temperature may be taken advantage of either to gain more capacity or to reduce steam consumption. This favorable consumption characteristic is not, of course, peculiar to steam jet compressors but has its parallel in the effect of suction temperature on the capacity of mechanical compressors. In the steam jet system, however, the flash tank introduces a very high thermal storage which makes it practicable to control the refrigerating machine directly, with little chance of hunting, in meeting load changes such as occur in air conditioning work.

In many systems, the chilled water is generated at a temperature as low as 40 F. In order to avoid the complicated systems of regulation which would be required if the refrigeration output were to be instantaneously changed, without lag or over travel, to meet the load changes, a thermostatically controlled valve bleeds the proper amount of chilled water into the dehumidifier circuit to produce the required dew-point. This procedure results in a practical but not particularly economical means of control, which has some understandable advantages from the production and installation standpoint.

The flywheel effect of the flash tank storage makes it possible to control the steam jet unit directly from the dew-point thermostat. This method of control will eliminate the cold water by-pass control and chilled water will be generated at the highest possible temperature compatible with the load.

Physically, the method of controlling the output of the machine is very simple. Usually, the capacity of the machine is divided among several boosters, one or more of which are turned on and off manually or automatically in accordance with the demand. The normal installation corresponds to a multiple installation of compressors having constant speed characteristics. If the thermostat operating the supply valves to the boosters is the dew-point thermostat, then the steam jet machine will operate only a sufficient time to maintain a flash tank temperature enough below the dew-point to meet the load. As the load increases, the dew-point will tend to rise, but will be restored by the lowering of the flash tank temperature caused by the operation of additional boosters. In this manner, with a fixed dew-point, the flash tank temperature will always be as high as the load permits and the machine will be operating on the point of greatest economy from the standpoint of this characteristic. If the greatest possible economy of operating energy is to be realized, not only must the steam jet machine be controlled by the dew-point directly but the dew-point itself should be maintained as high as possible.

It has been shown that because a steam jet refrigeration machine operates as a constant volume low head pump and because the volume of 1 lb of water vapor decreases rapidly as the saturation temperature or pressure rises without material change in latent heat per pound, it is possible, by raising the saturation or flash tank temperature, to substantially increase the capacity of the machine or to substantially reduce the steam consumption per ton of refrigeration produced. It has been indicated that the thermal storage introduced into the cycle by the supply of chilled water in the flash tank makes it practicable to control the flash tank temperature directly from the dew-point thermostat in a simple manner. The inescapable conclusion is, therefore, that operating energy economy will be greatly improved by operating at the highest possible flash tank temperature. Several specific cases have been discussed to illustrate possible variations of the operating scheme, which will make it possible to take the maximum advantage of the economy of flash tank temperature operation by showing the limitations of fixed dew-point temperature operation and the greater range of load character which can be covered at high dew-points without reheat. With reheat the ranges and economies possible are similar to those obtained without reheat. The authors believe that desired conditions can be obtained most satisfactorily over a wide range by allowing the dew-point to vary under control. The point with respect to steam jet is that whereas the rise of reheat and refrigeration may be set by operating conditions, the rise of saturation or flash tank temperature may be used to cut down the refrigerating energy required by the system over that required by constant flash tank temperature operation. By assuming as a limiting condition that all the air passed through the washer is outside air, it is seen that as the percentage of outside air is increased the offset to in-

creased operating costs by generating chilled water at the highest possible temperature may be considerable.

Steam Supply Pressure

If the pressure of the steam supplied to an operating booster on a steam jet machine is continuously reduced by throttling from the maximum, a point is reached where the jet breaks and refrigeration ceases. A jet will break at pressures which vary with the condenser vacuum, the break pressure decreasing as the condenser vacuum rises. A jet will be stable as long as the steam supply pressure is above the break pressure. Because the steam consumption of a jet is proportional to the initial pressure, it is obvious that to operate at any pressure materially above the break point is wasteful of steam, and any control that will maintain the supply pressure only slightly above the break point as the condenser vacuum changes will greatly reduce the energy required over that at constant pressure.

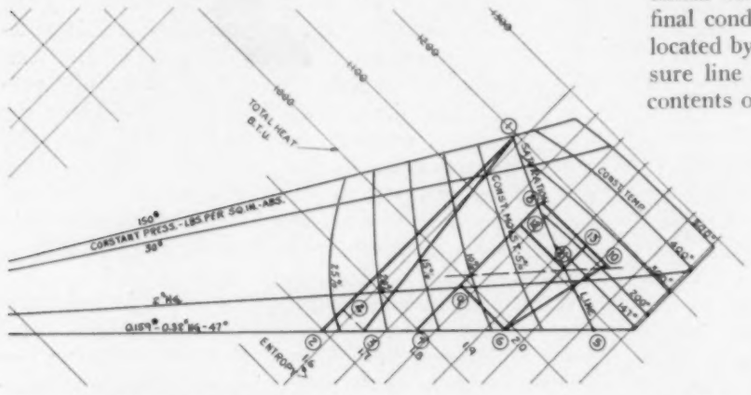

Fig. 10—Analysis of thermo-dynamic processes in booster of steam jet machine

To properly develop the technical and practical aspects of this characteristic of a steam jet refrigerating machine, it is advisable to digress slightly to indicate how the steam jet design problem may be analyzed with the aid of a Mollier diagram.

While the following simple analysis of the thermodynamic processes in the main booster is not basically new, it may be advanced to derive an understanding of a basic framework around which jets may be designed. The Mollier diagram, Fig. 10, shows the analysis used. Because certain of the condition points are determined chiefly by experiment, it should be realized that the analysis is more of a rational method of attack than a complete solution. The line 1-2 represents the isentropic expansion line in the high pressure steam nozzle, while line 1-3 represents the approximate actual expansion line. The energy available for propulsion is measured along the isentropic and is represented by the line 1-4. The two expansion lines, the isentropic (1-4) and the actual (1-3), are used to fix the contour of the main nozzles much as turbine nozzle contours are fixed. The initial condition of the flashed steam or refrigeration vapor may be represented by the point 5 on the saturation line. This flashed steam is mixed with the propelling steam from the jets at the pressure to which the jet steam has expanded. Assuming that the mixing of the two steams is accomplished without losses, the mixture may be represented by point 7 which indicates the increase in unconverted heat content of the jet steam by the admixture of flashed steam. To compress this mixture, available energy is represented by the isentropic compression line 7-8, per pound of mixture. Obviously, some of this energy will be lost in shock losses, etc., before compression starts and that part of the available energy lost may be represented by that portion of the isentropic between points 7 and 9. On the Mollier diagram, this loss can be represented as a throttling process from point 9 to point 6, giving the initial condition prior to compression as point 6. The final condition may be represented by point 10 which is located by considering that it is on the condensation pressure line and that it has a total heat equal to the heat contents of the propelling steam and flashed steam. Line 6-10 represents the actual path of the mixture compression. Line 11-13 is the available energy lost in compression, friction, shock and leaving loss. Thus, the total available energy of propulsion is accounted for. The condition of the steam at any time during compression is given by the line 6-10. The velocities at points in this path are given by the heat drop along the isentropic 13-6 from 13 to the point in question. The compressor cross-section contour may be calculated from these data.

The chart illustrates the energy changes taking place in a booster when the propelling steam acquires a high velocity by expanding to the flash tank pressure, mixes with the flash tank steam and then reconverts some of its velocity energy into static pressure as it pumps the flash tank vapors up to the condensing pressure. It is obvious, after a short examination of this chart, that the energy equations will indicate a lower required steam consumption if the line 6-10 is shortened by raising the flash tank temperature indicated by point 6. Similarly, it is obvious that an improvement in condenser vacuum unaccompanied by other changes will decrease require-

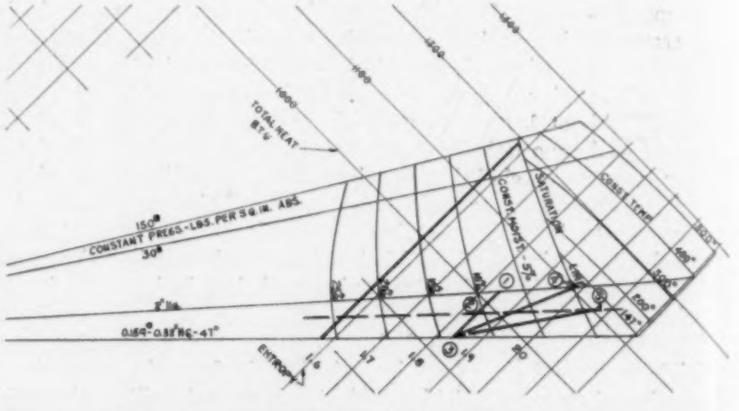

Fig. 11—Analysis of theoretical work of compression at constant supply pressure and changing condenser pressure

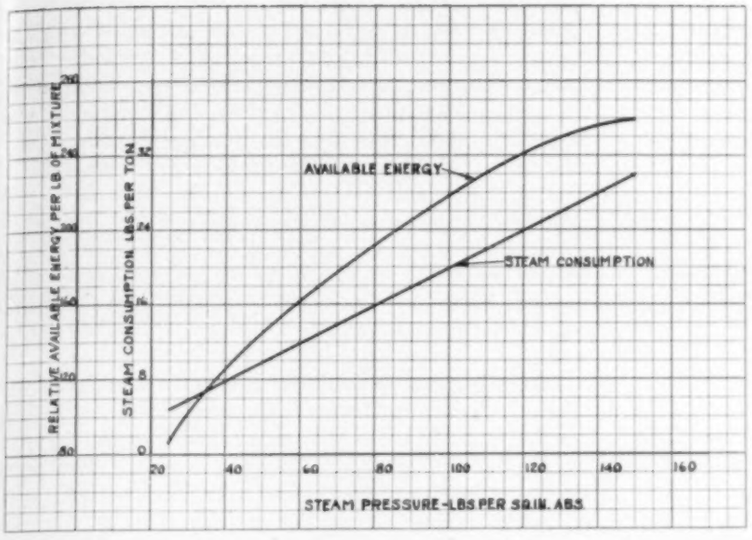

Fig. 12—Variation of available energy for propulsion and steam consumption with steam pressure

ments for propelling energy represented by the line 6-10 without decreasing the available amount of such energy. This condition of decreased efficiency is clearly shown on Fig. 11. Considering Fig. 11, it has been previously indicated that the useful work of compression per pound of a mixture at any point along the condenser pressure line is given by an isentropic such as 1-3. Suppose, however, that the condensation takes place at a higher vacuum because the cooling tower is operating under lower outside wet-bulb conditions, which results in condensation along a line of lower pressure. The useful work of compression is now 3-2. The actual compression line is seen to be 3-5 which is much worse than the former lines 3-4. From this, it is seen that the constant pressure operation is done at a considerable sacrifice of real efficiency at partial condenser loads. The explanation rests in the fact that the position of point 3 can not change as the ratio of propelling steam to flash tank steam is held constant by constant steam supply pressure. The result is that the efficiency, represented by the angle between the actual compression line and the isentropic, must vary with the condenser vacuum and grow worse as the vacuum improves.

It is possible to correct for this change in condenser vacuum and operate at practically constant thermodynamic efficiency by throttling the steam supply pressure. A throttled steam supply pressure causes the throttle flow and consequently the available energy for propulsion to decrease. Considering the jet as a constant volume compressor, the refrigeration tonnage can be assumed to remain constant while the steam supply pressure is being throttled. Tests of actual machines have shown that the steam supply pressure can be throttled to any point above the break point without appreciable capacity change. Actually, this constancy of capacity is probably due to the fact that the effect of the several changing conditions on the capacity adds up to zero. Thus, the nozzle expansion ratio changes and the nozzle efficiency falls off considerably when the steam supply pressure is varied widely. On the other hand, the shock losses and all velocity functional losses decrease as the initial pressure is lowered, so that in practice, it has been found that the jet acts as though its overall thermodynamic efficiency was fairly constant through a considerable range of operating pressure. The effects of a decreased steam flow on each part of the system are beneficial. Thus the cooling tower load decreases until equilibrium approaches and the tower gradually increases its ability to dissipate heat as it comes down on the temperature scale. On the Mollier diagram the effect of throttling the steam supply pressure is to move point 3 along the flash tank pressure line, in such a manner that the actual expansion line for the changed condenser vacuum remains parallel to the design expansion line and no decrease in efficiency takes place.

A measure of the gain made by decreasing the steam consumption by throttling the steam supply pressure is indicated in Figs. 12 and 13. On Fig. 12 is plotted as a straight line, the variation in the steam consumption of a fixed nozzle as the supply is adiabatically throttled from the initial design condition of 150 lb per square inch absolute, initially dry. If for each steam flow and throttled pressure, the theoretical energy made available by throttling and then expanding isentropically to the flash tank is calculated, a curve of available energy for propulsion results which may be drawn on the same chart. This curve indicates, relatively, the energy available for the propulsion of a pound of the mixture of propelling and flash tank steam to any condenser pressure. If more energy is required than is available, the jet will break, but if less is required than is available, that difference is wasted. The energy required, per pound of mixture, is treated as directly proportional to the theoretical work of compression from the flash tank to the condenser pressure at constant efficiency.

On Fig. 13, the relative work done on the flashed steam is plotted against condenser pressure. The work required to compress from a flash tank temperature of 46 F to a condenser pressure of 2 in. of mercury is taken

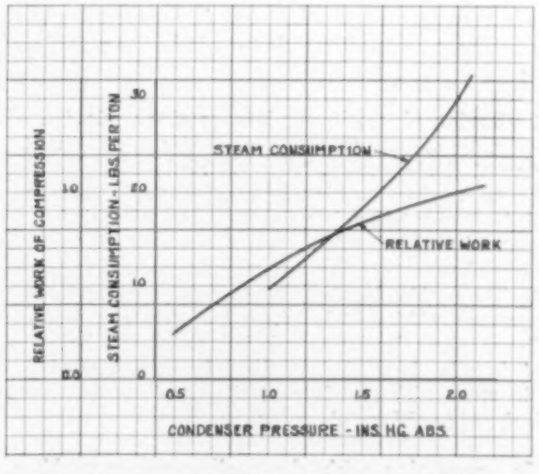

Fig. 13—Variation of relative work of compression and steam consumption with steam pressure

as unity. By combining the effect of the curve of relative available energy and relative work required, the effect of throttling the steam supply pressure on steam consumption can be indicated. The basic booster steam consumption used was 30 lb per ton of refrigeration, with a flash tank temperature of 46 F and a condenser pressure of 2 in. of mercury. Relatively, the steam consumption will be reduced from 30 to 10 lb per ton when the condenser vacuum rises from 28 to 29 in.

Cooling Tower and Evaporative Condensers

The cooling tower should be considered as the mechanism by which the heat rejected in the refrigeration cycle is finally disposed of to the atmosphere. It may be likened in theory to a large wet-bulb thermometer, the temperature of which is the result of the equilibrium between the gain of heat by the thermometer bulb due to dry-bulb temperature and the loss of heat due to water vaporization from the bulb. The ratio of heat transmission by conduction to that by evaporation is approximately constant over a considerable range. The cooling tower has been analyzed by considering it as a device which operates by virtue of the difference in vapor pressure between the water and the air along the path of cooling. An analogy between temperature difference and vapor pressure difference results from this treatment so that a tower may be assigned a coefficient of heat transmission which dimensionally is Btu per hour per cubic foot per inch of mercury integrated cooling potential.

Since the vapor pressure of water rises rapidly as the temperature of the water rises, the cooling tower is a device which will reject increasing quantities of heat as the water temperature increases, so that the equilibrium between it and the refrigerating machine must always be theoretically attained. Practically the condensing water temperature may not exceed the vacuum conditions under which the jet is stable. Towers are ordinarily designed so that the heat load specified will be rejected at a certain inlet water temperature and outside wet-bulb. The wet-bulb temperature seems to be fixed at 75 F. There are a small number of hours during the year in which the wet-bulb in New York City exceeds 75 F. During these hours when refrigeration is most needed, the machine functioning with a water tower temperature of 75 F will have operating difficulties. There are at least three solutions available which are:

1. The loss of capacity may be accepted at overload as must be done on mechanical compressor installations. This has the psychological disadvantage of being tangibly in evidence.
2. An auxiliary supply of cooling water for handling over peak loads may be cut into the tower circuit. This has the advantage of cutting installation costs at a negligible cost of peak water makeup.
3. The machine itself may be designed for a condensing water temperature which will be adequate to meet any external wet-bulb condition. In this case, it is imperative to include partial load throttling in the design. Since the writers believe that this is the most satisfactory from an all around economy standpoint, this should be elaborated.

In connection with the cooling water problem, the writers have available test data on a 68 ton steam jet unit operating in New York City with an evaporative condenser. These data are of interest because of the pioneer character of this evaporative condenser and steam jet combination on an air conditioning application. On a day when the outside wet-bulb temperature was 75.4 F the tests show that the machine developed 66.7 tons of refrigeration with a flash tank temperature of 45.7 F. The unit steam consumption was 27 lb per hour per ton of refrigeration.

Since the units are specifically designed and manufactured at present for each installation, there exists the opportunity to engineer more advantages into the installation than may be possible by using off the shelf equipment.

Steam jet coolers need suffer no operating handicaps when compared with other types of equipment. The starting of these machines may be made completely automatic with a single push button operation. They may be controlled for standby operation in connection with adiabatic cooling during the variable months. In this connection, the storage capacity of these machines gives a supply of cold water which may be drawn on for instant use. It is of interest to note that, generally, adiabatic cooling should have a prominent place in any economic study of coil versus air washer installations.

Controls and Requirements

It is quite feasible to consider the completely automatic operation of a steam jet machine and following this procedure through step by step seems a desirable approach to the basic control requirements.

1. Push button starts the cooling water circuit including pumps, towers and fans.
2. Pump circuit is interlocked with motor valve on secondary ejectors.
3. Condensate pump is started by a float switch, the pump output being controlled by a level controller.
4. When the condenser vacuum rises to the proper value, a vacuum switch energizes the main booster control circuit and starts the chilled water pump. The machine should now be on the line.
5. When the chilled water temperature reaches a predetermined setting, a thermostat throws in the throttling control.

The only safety device required practically is one thermostat. It throws the machine off the line should any failure take place which results in loss of load. This consists of a thermostat in the chilled water line set to shut the steam off at an abnormal chilled water temperature. No manual supervision in excess of that required with any other type of apparatus should be needed.

Economy Control

The steam pressure should be throttled in accordance with the condensing pressure so that the tonnage may be delivered at the lowest steam pressure compatible with the jet stability.

One apparatus developed to accomplish this consists of a valve in the steam line controlled by an air operated regulator which automatically balances the pressure differential existing between the condenser and the flash tank against steam pressure, so that every condenser vacuum condition calls for and maintains a proper oper-

ating steam pressure. Being a differential device, barometric changes do not affect it.

A modification of this principle has been successfully employed to control condensing water on purchased water supply. In this application, the steam pressure is held constant and the condensing water is throttled to maintain a constant differential between flash tank and condenser.

It is obvious that two courses are available to the designer of the steam jet in providing maximum economy and ability to handle peak load. It may be shown that certain relationships exist which make it possible to do this without material sacrifice of economy.

1. The stability of geometrically similar jets is proportional to the relationship between the propelling steam and the flashed steam.
2. The stability of the same jet is proportional to the steam pressure.
3. The flow of steam is proportional in the same jet, to the steam pressure.

Two courses are open. In most installations the designer of the machine has considered only constant pressure operation. Consequently, the boosters are designed to give the least steam consumption at the stated condensing water temperature. This has meant designing for the maximum available steam pressure. The result is, and has been, unfortunate. The machines do not perform at peak wet-bulbs without assistance.

On the other hand, if the designer initially gave the machine stability, safety in excess of that to be encountered under peak load, and then provided for throttling of the steam pressure to that required by the actual characteristics of the steam jet, the machine would perform at peak wet-bulbs.

Practical Results of Throttling Steam Supply Pressure

Studies of the consumption of steam to be expected over the cooling season in New York have shown that constant pressure installations will use, on the average, about 0.6 of the full load steam consumption. The seasonal steam consumption is the product of the following factors:

Installed Capacity (Tons) × Unit Steam Consumption (Pounds per Ton) × Hours of Operation × 0.6 = Seasonal Consumption.

Considering two installations which were both altered from constant pressure operation to variable pressure operation, it was found that the factor 0.6 was reduced to 0.4 or an indicated saving of one third in seasonal steam consumption.

Conclusion

The writers believe that the demonstration of the influence of flash tank temperature and steam supply pressure on economy is conclusive and is substantiated by performance records. Where a central station steam supply is available in Manhattan, the operating costs of steam jet are such that it can not be disregarded in any economic set-up. Nor can its simplicity and adaptability to simple control fail to appeal on an engineering basis. Until the present time, concentration has been stressed on the economic production of refrigeration, but in this paper discussion has been extended to the effective and economic use of refrigeration in order that users or potential users of air conditioning may correctly interpret the effect of the dew-point relation they select on the effectiveness and the operating costs of their installation.

Recommended Bibliography

General

1. Economic Application of Steam Jet Units, by M. A. Nelson (*Proceedings, N. D. H. A.*, 1936). Gives a background and general exposition of steam jet machines. Well illustrated.

Control

1. Operating Economies Reached by Control of Steam Jet Refrigeration, by A. R. Mumford and Ralph Menke (*Proceedings, N. D. H. A.*, 1935). Steam jet refrigeration discussed from the standpoint of automatic control of water temperature and steam pressure with theoretical and actual performance data. Description of controls.
2. Steam Jet Refrigeration Control by Water Flow Measurement, by J. M. Sperzel (*Proceedings, N. D. H. A.*, 1936). Gives test results and discussion of correlating control on air washers and refrigerating machine. Gives curves of energy, load and wet-bulb duration for New York City.
3. Improving Performance of Steam Jet Coolers, by Charles Hull (*Proceedings, N. D. H. A.*, 1937). Gives carefully compiled consumption records of 100 ton installation using differential condenser water control.

Testing

1. Testing of Steam Jet Coolers under Artificial Load, by A. A. Markson and Ralph Menke (*Proceedings, N. D. H. A.*, 1936). Outlines methods and results obtained by using artificial load technique for field tests. Gives consumption and stability characteristics of 100 ton machine.

Effect of Yaw on Vane Anemometers

Customary practice in using a vane anemometer is to mount the instrument so that the axis of rotation of the vane wheel is parallel to the axis of the fan or duct for which measurements are to be made. Generally, no attempt is made to determine whether or not the air flow is parallel to the duct or fan axis, although directional effects due to yaw may introduce errors of the order of one per cent for angles of instrument inclination less than approximately 20 deg.

Accurate results with an anemometer are only possible when mounted with its axis parallel to the fan axis and providing a correction factor for air stream inclination is known and applied. To establish suitable correction factors an investigation was undertaken to determine the effect of air stream inclination, in yaw, on the performance of three vane anemometers used in fan tests. Complete details concerning this investigation are reported in the Research Paper RP1056, released by the National Bureau of Standards December 1937.

Results show the presence of substantial directional errors within the 0 to 35 deg range of yaw angles. The several tests were performed in a wind tunnel in which a maximum directional error of about 5.5 per cent occurred when one of the instruments was displaced about 15 deg from alignment with the air stream of the wind tunnel. An error of about one per cent was observed when an anemometer of substantially the same diameter but of different design was displaced the same amount.

Proceedings of the 44th Annual Meeting

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

New York, N. Y.—January 24-28, 1938

D. S. Boyden, Boston, Mass.
President—1937

ESTABLISHING another record of attendance of over 1,000 individuals from all parts of the country and abroad, the 44th Annual Meeting of the Society, held at the Hotel Biltmore, New York, N. Y., January 24 to 28, was declared an outstanding success. All of the technical sessions were well attended, with especially large numbers present at the joint sessions held with the *National Warm Air Heating and Air Conditioning Association* and the *American Society of Refrigerating Engineers*.

The Council held two meetings during the week and assisted in the opening of the Fifth International Heating and Ventilating Exposition on January 24, which attracted nearly 45,000 people. Representatives from all of the local chapters were present at a conference on January 24 where Society officers and delegates discussed the future activities of the organization as related to local interests.

Numerous technical committee meetings were scheduled where research problems of common interest to the profession were outlined for future consideration. Codes relating to Minimum Requirements for Comfort Air Conditioning and Testing Stoker-Fired Heating Boilers were reported to the Society.

FIRST SESSION—

Tuesday, January 25, 1938, 10:00 A. M.

Pres. D. S. Boyden, Boston, Mass., called the 44th Annual Meeting of the Society to order in the ballroom of the Hotel Biltmore and introduced W. E. Heibel, New York, president of the New York Chapter, who welcomed the members and outlined briefly the program of entertainment and important events of the meeting. On behalf of the Society President Boyden responded and thanked Mr. Heibel and the New York Chapter for their cordial greeting.

President Boyden, in reporting the progress during the current year, referred the members to the detailed activities of the Society published in both the JOURNAL and TRANSACTIONS. He then pointed out the rapid growth that has taken place in the industry during the past year and particularly that branch referred to as *air conditioning*. Reference to the business conditions throughout the country during the last year were emphasized by the president as being better than for several years, which has contributed to the success of the Society. During the past year the Speakers Bureau of the Society was established, which enabled each Chapter to have various outstanding authorities deliver talks at the local Chapter meetings. According to President Boyden, this service was being used to considerable extent and was being well received. Another outstanding accomplishment of the Society during the past year was the establishment of closer cooperation between the heating, ventilating and air conditioning profession and those associated with the medical and health viewpoints. Much work along this line has already been accomplished by the Society, as pointed out by President Boyden, and there is evidence to indicate that the efforts of this cooperative work will be responsible for the improved health and longevity of mankind.

The Report of the Membership Committee was then presented by R. C. Bolsinger, Conshohocken, Pa.

Report of Membership Committee

Although it is the general thought that a Society like ours should not expand too rapidly, the increased interest in automatic heating, ventilating, and air-conditioning demands that we invite those associated, trained, and experienced in these arts to

E. Holt Gurney, Toronto, Canada
***President*—1938**

join our organization and to become active members so that the Society as a whole can benefit from their knowledge. Having these ideas in mind, your committee made an extensive drive, the results of which were very gratifying as shown in the accompanying schedule.

Number of Applications Received from January 15, 1937 to January 15, 1938	895
Number of Men Elected in 1937	473
*Number of Applications Pending	447
Number of Members Now on Rolls According to Grades (This group includes 49 Members who were recently Elected)	
Members	1466
Associates	692
Juniors	400
Students	35
Life Members	53
Honorary Members	2
Total	2648

Your committee has the following recommendations to make:

1. As the Chapters are the life-blood of the Society, additional Chapters should be formed wherever and whenever the membership is sufficient.
2. The problem of student membership should be considered so as to prove attractive to those who are really interested in heating, ventilating, and air-conditioning as we believe this to be for the good of the Society.

The committee takes this opportunity to thank the various Chapter membership committees for their very fine cooperation and the Past Presidents, W. H. Driscoll, W. T. Jones, S. R. Lewis, W. H. Carrier, A. C. Willard, H. M. Hart, F. B. Rowley, and J. F. Hale for their contributions regarding membership in the Society.

Respectfully submitted,

MEMBERSHIP COMMITTEE
R. C. BOLSINGER, *Chairman*
ALBERT BUENGER
W. A. RUSSELL

*The total number of applications received and elected during 1937 is not the same due to the fact that a few of the men elected in 1937 sent in their applications late in 1936.

A. V. Hutchinson, New York, Secretary, read the Report of the Council.

Report of the Council

The organization meeting of the Council was held in St. Louis January 27, 1937 with Pres. D. S. Boyden presiding, and during the year five meetings were called for the transaction of business. Announcement of the personnel of Standing and Special Committees was made, depositories for funds were selected, a budget was adopted and the appointment of A. V. Hutchinson, as Secretary, was made.

During the year meetings of the Council were held in St. Louis, Washington, D. C., Swampscott, Philadelphia and New York City.

The Council voted to hold the Semi-Annual Meeting 1938 at Hot Springs, Va., June 20-22, and the 45th Annual Meeting in Pittsburgh, Pa., January 23-25, 1939.

Charters to organize local chapters were approved for the Atlanta Chapter, Golden Gate Chapter at San Francisco, and Iowa-Nebraska Chapter.

A special Committee drafted rules of procedure for organizing local chapters and the Council adopted these regulations.

Suggestions for Amendments of the Society's Constitution and By-Laws were received and the Council approved them for submission to members.

The Council voted that dues for 1938 would be $18 for Members and Associates, $10 for Juniors, $3.00 for Students with initiation fees of $10 for candidates eligible for Member and Associate grades and $5 for those eligible for Junior grade. Approval was given to the transfer of $4100 to the Research Fund in addition to the allotment from membership dues.

Approval was given to the request of the Guide Publication Committee to change the name of THE A. S. H. V. E. GUIDE to THE HEATING, VENTILATING, AIR CONDITIONING GUIDE.

The Council considered the recommendations of a Special Committee which had studied the needs of local chapters and voted to establish a Speakers Bureau. It was voted to furnish each chapter upon request, speakers for not more than two meetings between the months of October and April, and also permit the Speakers Bureau to furnish additional speakers when chapters would defray the expense involved.

It was voted that one delegate from each Chapter be brought to the 1938 Annual Meeting and the round trip railroad expense be furnished by the Society.

During the year 16 life memberships were conferred upon the following men: F. C. Black, B. C. Davis, J. A. Donnelly, E. J. Febrey, R. H. Feltwell, R. L. Gifford, J. F. Hale, J. G. Hayes, J. D. Hoffman, H. G. d'Issertelle, J. H. Kitchen, Walter Leck, H. C. Meyer Jr., F. W. Powers, J. N. Russell and G. J. Tobin.

Approval was given by the Council to the revisions sponsored by 17 engineering societies on the Model Registration Law for Professional Engineers and Land Surveyors.

Formal action was taken in accordance with the Constitution and By-Laws on 473 membership applications of which 15 were reinstatements, 48 resignations, and 176 cancellations.

Respectfully submitted,

THE COUNCIL.

The Report of the Secretary was presented by Mr. Hutchinson.

Report of the Secretary

To give a detailed description of the activities carried on at the headquarters office by the Secretary and the staff of eight during the past 12 months would require reams of paper so that this report will be confined to a summary of the most important items.

The year 1937 was an unusually successful one for the A.S.H.V.E. because of a phenomenal increase in membership,

the organization of chapter activities in three major cities and the extension of research investigations into wider fields.

The volume of work carried on at the headquarters office exceeded any previous year in the history of the Society, because more copies of THE GUIDE were distributed, monthly JOURNALS were larger, two volumes of TRANSACTIONS were issued, and two codes were completed, in addition to the normal routine of handling membership applications, changes of address, employment and other administrative details.

The employment service work has been carried on throughout the year with a notable increase in volume during the last few weeks. Through the chapters, efforts have been made to list those needing employment as rapidly as their needs were made known.

Up to the middle of December, it was evident that 1937 would show a greater increase in membership than had ever been recorded previously. During the last two weeks of the year, the number of applications received almost reached 50 per cent of the total filed during the first 11 months. The total number of applications for the year is 895. On January 1, 1937 there were 2372 members on the rolls and during the year 473 new members were elected; losses were 197, so that the year ended with 2648 members or a net gain of 272.

A heavy addition to the office work was imposed by the establishment of the Speakers Bureau to assist the chapters and from September to December arrangements were completed for 17 speakers to address 19 Chapters.

For those who are statistically minded, it might be noted that over 100,000 letters were mailed and if the stamps purchased were placed side by side they would reach from the New York

BALANCE SHEET*

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y.
December 31, 1937

ASSETS

SOCIETY			
CASH—			
On deposit	$57,746.69		
On hand	100.00		
On hand for deposit	42.95		
		$57,889.64	
In closed bank		457.05	
			$58,346.69
INVESTMENTS (AT COST)			
Securities (Market value $11,544.00)		$11,525.31	
Add—Accrued interest		61.46	
			11,586.77
ACCOUNTS RECEIVABLE			
Membership dues	$13,700.06		
Less—Reserve for doubtful	8,315.27		
		$ 5,384.79	
Advertisers and sundry debtors	$33,631.54		
Less—Reserve for doubtful	1,521.12		
		32,110.42	
			37,495.21
INVENTORIES			
TRANSACTIONS		$ 3,255.67	
Emblems and certificate frames		34.16	
			3,289.83
PERMANENT			
Library		$ 300.00	
Furniture and fixtures	$ 6,056.66		
Less—Reserve for depreciation	5,080.92		
		975.74	
			1,275.74
DEFERRED CHARGES			
Meeting expenses			216.73
			$112,210.97
SPECIFIC FUNDS			
ENDOWMENT FUND			
Cash on deposit			$ 5,322.52
Securities at cost (Market value $8,140.00)		$11,918.65	
Add—Accrued interest		111.35	
			12,030.00
			17,352.52
F. PAUL ANDERSON AWARD FUND			
Cash on deposit			1,164.61
RESEARCH FUND			
ENDOWMENT FUND			
Cash on deposit		$ 491.90	
Cash in closed bank		183.03	
		$ 674.93	
UNRESTRICTED FUND			
Cash on deposit		$ 9,171.05	
Cash on hand		50.00	
Cash on hand for deposit		2,385.33	
		11,606.38	
			12,281.31
			$143,009.41

LIABILITIES AND CAPITAL

SOCIETY			
ACCOUNTS PAYABLE			$15,696.27
PAYABLE TO RESEARCH FUND			
On dues as and when collected			3,318.74
ACCRUED ACCOUNTS			
Compensation—Secretary and staff			2,579.79
RESERVE FOR PUBLICATION			
TRANSACTIONS, Vol. 43			3,500.00
DEFERRED INCOME			
Prepaid dues—Members		$ 1,396.16	
Prepaid dues—Proposed members		2,134.55	
			3,530.71
RESERVE FOR ENDOWMENT FUND REPLENISHMENT			7,500.00
TOTAL LIABILITIES			$36,125.51
GENERAL FUND			
Society			76,085.46
			$112,210.97
SPECIFIC FUNDS			
ENDOWMENT FUND			
Principal			$16,363.28
Unexpended income			989.24
			17,352.52
F. PAUL ANDERSON AWARD FUND			
Principal			$ 1,000.00
Unexpended income			164.61
			1,164.61
RESEARCH FUND			
ENDOWMENT FUND			
Principal		$ 600.00	
Unexpended income		74.93	
			$ 674.93
UNRESTRICTED FUND			
Balance—January 1, 1937		$15,060.16	
Excess of expenses over income for the year ended December 31, 1937		3,453.78	
		11,606.38	
			12,281.31
			$143,009.41

NOTE "A" This Balance Sheet is subject to comments contained in the letter attached to and forming a part of this report.

*From report of Tusa & LaBella, Certified Public Accountants.

to New Jersey portals of the Lincoln Tunnel. The number of GUIDES shipped amounted to 30 tons or a car load and one half. The proof reading of the text section of THE GUIDE involved 350,000 words equivalent to four full length novels. The number of GUIDE orders handled amounted to 8000 items. Address changes numbered 1450 and checks issued totaled 725.

During the year assistance was given in the organization of three new chapters with headquarters at Atlanta, Ga., San Francisco, Calif. and Omaha, Nebr.

Your Secretary assisted in the organization of two of these Chapters and visited seven other chapters during the year and also attended the meetings of numerous national organizations interested in A.S.H.V.E. activities.

The reports of the many committees to be presented at this meeting will give an accurate picture of the varied activities of the Society.

It is a great pleasure to acknowledge the cooperation given by the Officers and Council, Chapter Officers and the Committee members, who have assisted in making 1937 a banner year for the Society.

It is fitting that special mention should be made of the splendid service rendered by Mr. James and the other members of the headquarters staff. Their ability to respond to the increased tempo of activities and to handle an extraordinary volume of work deserves special recognition by the members.

Respectfully submitted,

A. V. HUTCHINSON, *Secretary.*

Report of the Finance Committee

J. F. McIntire, Detroit, chairman of the Finance Committee, gave the Report of the Society's Certified Public Accountant, showing the conditions of the finances for the calendar year 1937. In commenting on the report, Mr. McIntire referred to the added income of the Society, which came largely from an increase in GUIDE advertising and sales. It was indicated that over 51 firms contributed to the Research Fund during the current year. The chairman of the Finance Committee emphasized the necessity of immediate action when members become delinquent in dues.

BUDGET COMPARISON

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
New York, N. Y

For the Calendar Year Ended December 31, 1937

	Actual	Budget Provision	Increases Decreases
Income:			
Membership dues	$ 23,167.40	$23,000.00	$ 167.40
Editorial contract	14,131.43	13,500.00	631.43
Profit—Sales of emblems and certificates	53.07	100.00	46.93
Profit—Reprints and books	309.71	300.00	9.71
Sales TRANSACTIONS	704.32	800.00	95.68
Interest—Savings banks and securities	973.07	1,000.00	26.93
GUIDE—Advertising	34,418.78	31,000.00	3,418.78
GUIDE—Copy sales	31,877.37	30,000.00	1,877.37
	$105,635.15	$99,700.00	$ 5,935.15
Expenses:			
General Society Activities:			
Meetings	$ 1,857.37	$ 1,600.00	$ 257.37
Chapter allowance for meetings	1,000.00	1,000.00	.....
Promotional	.....	3,000.00	3,000.00
Speakers Bureau	849.28	1,000.00	150.72
Subscriptions (H.P. & A.C.)	5,058.79	5,000.00	58.79
Postage	2,393.45	2,500.00	106.55
Rent	2,589.96	3,000.00	410.04
TRANSACTIONS	3,500.00	3,500.00	.....
Codes (Net of sales)	139.76	500.00	360.24
Salaries (Includes additional compensation)	17,670.15	19,000.00	1,329.85
General printing	911.93	800.00	111.93
Membership certificates and emblems	288.10	600.00	311.90
President's fund	1,145.63	1,800.00	654.37
Travel—Secretary	1,291.45	1,500.00	208.55
Council and chapter delegates travel	3,497.25	4,400.00	902.75
Bank charges and discounts	55.72	100.00	44.28
Multigraphing	635.38	500.00	135.38
Telephone	641.37	500.00	141.37
Telegraph	306.38	300.00	6.38
Professional services	400.00	500.00	100.00
General office	736.72	600.00	136.72
Office supplies	269.02	500.00	230.98
Furniture and fixtures	201.55	500.00	298.45
Special appropriations	998.27	550.00	448.27
	46,437.53	53,250.00	6,812.47
Cost of GUIDE:			
Editorial and advertising salaries	6,286.64	7,200.00	913.36
Committee travel	251.01	750.00	498.99
Printing and binding	15,829.26	18,500.00	2,670.74
Mailing	1,575.00	2,000.00	425.00
Paper	4,546.96	4,000.00	546.96
Cuts	514.32	500.00	14.32
Advertising sales promotion	1,452.61	2,000.00	547.39
Copy sales	7,391.70	7,500.00	108.30
Yearbook section	1,250.11	1,000.00	250.11
	39,097.61	43,450.00	4,352.39
	$ 85,535.14	$96,700.00	$11,164.86
Unbudgeted — Appropriations to research fund	4,100.00	.....	4,100.00
	$ 89,635.14	$96,700.00	$ 7,064.86

BUDGET COMPARISON—RESEARCH FUND

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
New York, N. Y

For the Calendar Year Ended December 31, 1937

	Actual	Budget Provision	Increases Decreases
Income:			
Balance from 1936 committee—Cash in Pitts. and New York banks	$15,060.16	$12,088.23	$2,971.93
Dues—Current and prior years	13,744.19	13,800.00	55.81
Interest on bank balances	.....	30.00	30.00
Interest on savings account	59.76	70.00	10.24
Interest on society endowment	.....	600.00	600.00
Contributions—General	5,200.00	10,000.00	4,800.00
Contributions—Earmarked	500.00	500.00	.....
Special council appropriation	2,500.00	4,000.00	1,500.00
Special council appropriation	200.00	200.00	.....
Special council appropriation	1,400.00	1,400.00	.....
Contributions—"Glass" study	3,650.00	4,075.00	425.00
	$42,314.11	$46,763.23	$4,449.12
Expenses:			
Travel—Committee and laboratory staff	$ 1,707.82	$ 2,000.00	$ 292.18
Salaries—Laboratory staff	13,296.00	15,000.00	1,704.00
Salaries—Student help	1,700.55	1,800.00	99.45
Cooperative research—Earmarked	731.25	731.25	.....
Cooperative research—General	2,074.22	5,400.00	3,325.78
Cooperative research—Medical school	2,025.00	2,100.00	75.00
Laboratory supplies and equipment	692.56	700.00	7.44
Office supplies and equipment	518.13	500.00	18.13
Printing	.....	400.00	400.00
Exhibits	.....	200.00	200.00
Correlating thermal research	1,260.00	1,500.00	240.00
Salary—Office of chairman	275.75	900.00	624.25
Promotion and publicity	5,091.16	4,400.00	691.16
Meetings	161.19	300.00	138.81
Contingency fund	.....	500.00	500.00
N. D. H. A. exhibit	211.03	200.00	11.03
Special study on "Glass"	963.07	2,125.00	1,161.93
	$30,707.73	$38,756.25	$8,048.52
UNAPPROPRIATED BALANCE	11,606.38	8,006.98	3,599.40
	$42,314.11	$46,763.23	$4,449.12

NOTE: The above Budget Comparison includes all revisions voted by the Committee on Research during the calendar year 1937. We note that in preparing the 1937 budget the Research Committee has shown the cash balance taken over from the 1936 Research Committee at $12,088.23 due to the exclusion of the cash on deposit with the Bank for Savings.

President Boyden announced the first technical paper and introduced A. R. Mumford, New York, who presented the paper, Application and Economy of Steam Jet Refrigeration to Air Conditioning, by A. R. Mumford and A. A. Markson. (Complete paper published in March, 1938, A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*) A written discussion was presented by E. H. Taze, New York, and M. A. Nelson added additional remarks, which were answered by the co-author, Mr. Markson.

The next technical paper, Cooling Tower Equipment and Its Relation to Water Conservation, was presented by the author, S. I. Rottmayer, Chicago, Ill. (Complete paper to be published in A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*) Written comments on this paper were presented by J. R. Hertzler, York, Pa., J. deB. Shepard, Baltimore, Md., and C. F. Kayan, New York. Oral comments were contributed by Mr. Mumford, Mr. Brown and Prof. W. H. Severns, Urbana, Ill., all of which were responded to by the author.

The Report of the Tellers was then presented by John James, New York, as follows:

Reports of Boards of Tellers

Ballot for Officers

	For
President—E. H. Gurney	700
First Vice-President—J. F. McIntire	702
Second Vice-President—F. E. Giesecke	702
Treasurer—A. J. Offner	702
Members of the Council—Three-Year Term	
N. D. Adams	702
A. P. Kratz	702
J. H. Walker	702
G. L. Wiggs	701
Total Ballots Counted	734
Ballots Invalid	32

Scattering votes were recorded for F. J. Linsenmeyer, F. C. Hibbs, and A. W. Luck.

Ballot for Committee on Research

Three-Year Term	
M. K. Fahnestock	702
W. L. Fleisher	700
R. J. Tenkonohy	702
T. H. Urdahl	701
B. M. Woods	702
Total Ballots Counted	734
Ballots Invalid	32

Scattering votes were recorded for Tom Brown and C. A. Thinn.

BOARD OF TELLERS
R. V. SAWHILL, *Chairman*
J. R. MURPHY
R. G. OLSON

Ballot for Amendment to Constitution and By-Laws Article B-X, Section 1, and Article B-XI, Section 1

Number of Ballots *For*	666
Number of Ballots *Against*	22
Total Number of Ballots	688
Illegal Ballots	33

BOARD OF TELLERS
M. C. BARNUM, *Chairman*
E. C. BLACKBURN, JR.
G. A. DORNHEIM

Ballot for Code of Minimum Requirements for Comfort Air Conditioning

Number of Ballots *For*	[illegible]
Number of Ballots *Against*	[illegible]
Total Number of Ballots	570
Illegal Ballots	55

BOARD OF TELLERS
C. L. CHASE, *Chairman*
C. S. PABST
C. D. SHIELDS

The session adjourned at 12:30 p. m.

SECOND SESSION—

Tuesday, January 25, 1938, 2:00 P. M.

A joint meeting with the *National Warm Air Heating and Air Conditioning Association* was held in the ballroom of the Hotel Biltmore, with Pres. D. S. Boyden and Pres. W. L. Rybolt, Ashland, Ohio, of *N.W.A.H.A.C.A.* presiding jointly as chairmen. In opening the session, President Boyden extended a word of welcome to the allied association, who were kind enough to arrange their annual meeting so that they could meet with the Society at the same time and place in order to take advantage of the technical papers, which were of special interest to both groups.

In responding to this word of welcome, President Rybolt indicated the pleasure of the association in being present at the joint meeting. He referred to the celebration of the association's 20th anniversary of research work and introduced F. G. Sedgwick, an active member of the research activities of the association. Through a cooperative agreement with the University of Illinois, Mr. Sedgwick indicated that the association had turned the light of research investigation on the major problems of the industry for the last 20 consecutive years. Reference was made to the close cooperation of the A.S.H.V.E. in connection with this program, where approximately a quarter of a million dollars had been invested at the University of Illinois in research activities. Although the association was responsible for the erection of the Research Residence at a cost of $20,000, Mr. Sedgwick indicated how the A.S.H.V.E. had helped to immortalize this project by cooperative research activity. Prof. A. P. Kratz, Urbana, was then introduced by Mr. Sedgwick and gave a talk on Fundamentals Developed from 20 Years of Research, which he presented with the assistance of numerous slides.

Upon the completion of Professor Kratz' remarks, President Boyden introduced the managing director of the *N.W.A.H.A.C.A.*, A. W. Williams, Columbus, Ohio. The president-elect, E. Holt Gurney, Toronto, Ont., of the A.S.H.V.E was introduced to the meeting and was asked to preside for the remainder of the technical session. Questions were directed to Professor Kratz by Prof. J. E. Nicholas, State College, Pa., B. F. McLouth, Lansing, Mich., G. S. Williams, New Haven, Conn., R. L. Davison, New York and E. H. Lloyd, Washington, D. C.

A paper entitled Air Distribution from Side Wall Outlets, by D. W. Nelson and D. J. Stewart, was given by Professor Nelson, Madison, Wis. (Complete paper to

be published in A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*) In the absence of written discussions, oral comments were presented by Prof. B. M. Woods, Berkeley, Calif., Prof. G. L. Tuve, Cleveland, Ohio, Prof. F. B. Rowley, Minneapolis, Minn., and Professor Nicholas, all of which were answered by the co-author, Mr. Stewart, Rockford, Ill.

H. Merwin Porter, Minneapolis, Minn., president of the *Heating, Piping and Air Conditioning Contractors National Association,* was introduced to the meeting and extended greetings from that organization.

The last technical paper, Condensation Within Walls, by F. B. Rowley, A. B. Algren and C. E. Lund, was presented by Professor Rowley. (Complete paper published in January, 1938, A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*) Written discussions submitted by L. A. Harding, Buffalo, and H. C. Bates, Corning, N. Y., were read by Mr. Williams. Other written discussions were presented by E. C. Lloyd, Lancaster, Pa., and Prof. E. R. Queer, State College, Pa. Oral comments were contributed by Mr. Davison, J. J. Foley, Toronto, L. K. Hughes, Toronto, Mr. McLouth, J. U. Maley, Youngstown, Ohio, H. L. Stevens, Hutchinson, Kan., and J. H. Bracken, Chicago, Ill., and were answered by Professor Rowley.

The session adjourned at 5:15 p. m.

THIRD SESSION—

Wednesday, January 26, 1938, 10:00 A. M.

President Boyden opened the session and first called for the Report of the Treasurer, presented by A. J. Offner, New York.

Report of the Treasurer

When one sees the many activities being carried on at these meetings, when one receives THE GUIDE and other publications, when one notes the excellent and important work being carried on by our Research Committee, one realizes that it not only takes much work but also much money. We would therefore like to appeal to the members to pay their dues promptly, not only so that the necessary money is available, but also in order that we do not have the unnecessary expense of billing such members.

We also have another problem and that is the collection of dues which are past due. While we appreciate and understand the reason in many cases, and while we try to be as lenient and considerate as we possibly can, we are forced by the requirements of our Constitution to drop such members. We are very reluctant to do so and we would also like to appeal to the help of the members in such cases.

W. A. Danielson, Fort Knox, Ky., chairman of the Committee on Research, was then introduced and abstracted the Report of the Committee on Research for the year 1937. (Complete report published in the January, 1938, A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*)

The first paper of the technical session, Studies on Bacterial Control in Air Conditioning, by T. S. Carswell, H. K. Nason and J. D. Fleming, was given by Mr. Nason, St. Louis, Mo. (Complete paper to be published in A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*) Written discussions were presented by V. T. Kartorie, York, Pa., H. C. Murphy, Louisville, Ky., and H. L. Baker, Jr., New York. Additional discussion was contributed by A. A. Adler, New York, F. R. Bichowsky, Ann Arbor, Mich., and J. J. Aeberly, Chicago, which were responded to by Mr. Nason.

The paper, Physiologic Response of Man to Environmental Temperature, by F. K. Hick, R. W. Keeton and Nathaniel Glickman, was presented by Dr. Hick. (Complete paper published in March 1938 A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*) As chairman of the Technical Advisory Committee on Relation of Body Changes to Air Changes, Dr. E. V. Hill, Chicago, stressed importance of the cooperative work of the Society which has been initiated at the University of Illinois Medical School, and referred to this paper as being the initial report of a series of papers which will be published in the future covering this work. Written discussions were then contributed by Mr. Aeberly, E. J. Rodee, New Haven, Conn., Prof. C.-E. A. Winslow, New Haven, Conn., Dr. C. A. Mills, Cincinnati, W. L. Fleisher, New York, and Prof. C. P. Yaglou, Boston. Additional comments were given by N. D. Adams, Rochester, Minn., and Professor Rowley, which were answered by Dr. Hick in closing.

Telegrams expressing their regret at not being able to attend the Annual Meeting were read from Dr. A. C. Willard, Urbana, Ill., and C. V. Haynes, Philadelphia, Pa., past presidents of the Society.

The last paper of the session, Control of Body Heat Loss Through Radiant Means, by C. A. Mills and Cordelia Ogle, was presented by Dr. Mills. (Complete paper published in November, 1937, A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*)

Written discussions were presented by T. N. Adlam, New York, and J. R. Parsons, New York, and additional comments were contributed by Professor Winslow, Mr. Fleisher, Anson Hayes, Middletown, Ohio, Mr. Shepard and Dr. Adler.

The session adjourned at 12:45 p. m.

FOURTH SESSION—

Wednesday, January 26, 1938, 2:15 P. M.

On Wednesday afternoon President Boyden called the Joint Session with the *American Society of Refrigerating Engineers,* held in the Hotel Biltmore, to order and introduced the presiding chairman, W. H. Carrier, Syracuse, N. Y., past president of both societies. Before presenting the first technical paper of the session, Mr. Carrier introduced the president of the *A.S.R.E.,* H. M. Williams, Dayton, Ohio.

The paper, Control of Air Conditioning in Large, Medium and Small Buildings, by W. E. Zieber and S. F. Nicholl, was given by Mr. Nicholl, who abstracted the paper, supplemented by lantern slides. Discussion was contributed by A. B. Newton, Minneapolis, Minn., A. B. Snavely, Hershey, Pa., and the closing remarks were added by the co-author, Mr. Zieber.

The next technical paper, Physiological Reactions and Sensations of Pleasantness Under Varying Atmospheric Conditions, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge, was ably given by Professor Winslow. (Complete paper published in January, 1938, A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*) One written discussion presented by Professor Yaglou was read, and additional remarks were made by

T. W. Carraway, Dallas, Tex., Mr. Aeberly and Mr. Fleisher.

The last paper of the joint session, Performance Tests of Asbestos Insulating Air Duct, by R. H. Heilman and R. A. MacArthur, was given by Mr. Heilman. (Complete paper published in February, 1938, A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*) R. R. Gannon, Cincinnati, Ohio, read a prepared discussion and others who contributed oral comments were Mr. Pierce, Mr. Carrier, Mr. Snavely, Mr. Lloyd and Professor Yaglou.

The gavel of authority was then passed back to President Boyden, who expressed the appreciation of the Society in being able to meet jointly with the *A.S.R.E.* He further stated that the excellent attendance indicated the mutual interest of the two societies and hoped that on future occasions similar meetings would be arranged.

Following the few brief announcements, the meeting adjourned at 4:20 p. m.

FIFTH SESSION—

Thursday, January 27, 1938, 10:15 A. M.

After calling the session to order, President Boyden introduced Walter Sturrock, who presented the paper entitled Effects of Artificial Lighting on Air Conditioning. (Complete paper published in February, 1938, A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*) Several written discussions were prepared for this paper and were presented by the contributors, W. F. Friend, New York, A. D. Cameron, New York, H. M. Sharp, Buffalo, W. R. Heath, Buffalo, C. M. Ashley, Syracuse, N. Y., and D. W. McLenegan, Bloomfield, N. J. Additional remarks were given by C. S. Woodside, Bloomfield, N. J., G. W. Martin, New York, and G. E. May, New Orleans, La.

The second paper of this technical session, Comparative Analysis of Office Building Air Conditioning Systems, by J. R. Hertzler, was given by the author. (Complete paper to be published in A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*)

The final paper of the session, Ventilating the Lincoln Vehicular Tunnel, was presented by C. W. Murdock. (Complete paper published in January, 1938, A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*) Past president of the Society A. C. Willard, former consultant on ventilation of the New York and New Jersey Bridge and Tunnel Commission, submitted a written discussion which was read by Mr. James. Prof. James Holt, Cambridge, Mass., questioned the author in regard to the design of the system, which was answered by Mr. Murdock.

After announcements pertaining to the inspection trips for the afternoon, the session adjourned at 12:15 p. m.

SIXTH SESSION—

Friday, January 28, 1938, 10:00 A. M.

The final technical session of the 44th Annual Meeting was called to order by President Boyden.

L. J. Fowler, London, England, was introduced to the meeting and conveyed the special and best wishes of the president and council of the *Institution of Heating and Ventilating Engineers of Great Britain* to the Society. He congratulated the organization on the successful year it had just completed and wished every success for the Annual Meeting. Mr. Fowler also conveyed to the president-elect, E. Holt Gurney, the sincere congratulations of the English Institution and expressed confidence that his year of office would be memorable both to him and to the Society. In closing, the speaker indicated how happy he was to come back and see so many friends that he made eight years ago when he was the joint author with A. H. Barker of an interesting paper on Radiant Heating, which was presented before the Society at that time.

The Report of the Committee on Code for Testing Stoker-Fired Steam-Heating Boilers was given by the chairman, C. E. Bronson, Kewanee, Ill.

Report of Committee on Code for Testing Stoker-Fired Steam-Heating Boilers

The Committee was assigned the duty of preparing a new Code for the Testing of Stoker-Fired Steam-Heating Boilers and of reviewing the present test codes to determine if revisions were essential.

In the Code for Testing Stoker-Fired Steam-Heating Boilers presented for your consideration, the general scheme of arrangement is similar to the Code for Testing Oil-Fired Boilers. Its provisions admit the burning of any type of solid fuel such as anthracite or bituminous coal. The type of stoker or boiler is not limited.

Particular attention is directed to the purpose of the Code, as expressed in the first sentence of the introduction. The Code is intended to provide a standard method for conducting and reporting tests and to determine heat efficiency and performance characteristics. It does not fix the conditions that must be fulfilled for the acceptance or approval of the combination of boiler and stoker. In other words, this Code does not seek to specify the coal burning rate of a stoker required for the proper performance of a given boiler, but merely to report the results obtained with any combination.

Results of tests may be reported in the standard or long form or in the short-test form, except that for intermittent tests only the short-test form is required.

The Committee gave due consideration to reporting heat balance for intermittent tests, but on account of the difficulty in obtaining accurate average values throughout the test period, did not consider its inclusion advisable.

Furnace volume and hearth or grate area are defined for the purpose of this Code. No attempt has been made to define either effective furnace volume or active grate area. It is realized that combustion may be more intense in some particular zone, yet the proposed definitions outline practical values which can be used as a basis for comparison.

In reviewing the present test codes, it was the consensus of opinion that revisions are desirable. Both the general test code designated as Codes 1 and 2, also the Code for Testing Oil-Fired Steam-Heating Boilers, should be changed to eliminate reference to evaporation in pounds of water per pound of fuel, using instead the more widely used heat unit basis. The lack of time did not permit a more detailed or specific report on the changes required, and furthermore the committee feels that this duty should perhaps devolve upon a new committee whose personnel would recognize those primarily interested in oil burning.

Test Codes have an important bearing on the heating and ventilating industry and annual revision may be necessary to maintain them in up-to-date manner. It is hoped that a Standing Committee will be assigned for this duty.

It was regularly moved and seconded that the Code be presented to the Society membership for letter ballot.

Before presenting the first technical paper of the ses-

sion, President Boyden introduced W. G. Case, London, England.

The paper, Draft Temperatures and Velocities in Relation to Skin Temperature and Feeling of Warmth, by F. C. Houghten, Carl Gutberlet and Edward Witkowski, was given by Mr. Houghten. (Complete paper published in February, 1938, A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*) Questions were directed to the author by Professor Tuve, Mr. Parsons, Mr. Aeberly and M. C. Beman, Buffalo, which were answered by Mr. Houghten.

Installation of Officers

The technical session was then interrupted for a short period for the installation of the newly elected officers, which was conducted by Past President Homer Addams.

The newly elected officers, who were installed, included Pres. E. Holt Gurney, Toronto, 1st Vice-Pres. J. F. McIntire, Detroit, 2nd Vice-Pres. F. E. Giesecke, College Station, Tex., and Treas. A. J. Offner, New York. The four new members of the Council, N. D. Adams, Rochester, Minn., J. H. Walker, Detroit, Prof. A. P. Kratz, Urbana, Ill., and G. L. Wiggs, Montreal, were escorted to the rostrum and introduced to the meeting.

President Gurney announced the next technical paper, Study of Methods of Control and Types of Registers as Affecting Temperature Variations in the Research Residence, by A. P. Kratz and S. Konzo, which was given by Mr. Konzo. (Complete paper published in December, 1937, A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*) Additional comment was made by Professor Kratz that, although the paper refers specifically to two-speed fan operation, the important points involved the use of varying volumes of air, which could be accomplished either by change in fan speed or by a system of damper arrangements.

The final technical paper of the Annual Meeting was presented by A. B. Newton, entitled Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley. (Complete paper published in December, 1937, A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning.*)

Upon completion of the presentation of the paper by Mr. Newton, President Gurney turned the meeting over to Vice-President McIntire, who presided for the balance of the session.

Additional comments pertaining to the paper were presented by Mr. Houghten and discussion was contributed by Mr. Parsons.

The Resolutions Committee's report was presented by J. H. Van Alsburg, Chicago.

Resolutions

On motion of J. H. Van Alsburg, seconded by T. H. Urdahl, it was voted that this expression of thanks and appreciation be adopted by the members attending the 44th Annual Meeting:

We have had a very wonderful Annual Meeting, which is only partially proven by the registration. This is one of the largest meetings we have ever had and total registration exceeds 1000. We are having the best and largest Exposition and we have many to thank for the fine work they have done in making this meeting such a wonderful success.

We wish to definitely record our appreciation of the efforts of those who individually and collectively have made this meeting outstanding in the history of the Society.

We, first of all, thank the New York Chapter and its Committee on Arrangements for the very fine work they have done in providing the social features which contributed so materially in making this meeting a success. Especially, commending them for the wonderful entertainment provided for the ladies.

We thank the management of the Hotel Biltmore for the many courtesies and exceptional services which they have rendered in providing a maximum of comfort during our stay.

We thank the press and trade publications for the fine publicity they have given the Society.

We thank the authors of papers who have contributed so largely to the interest in attendance at our technical sessions.

We express our sincerest appreciation to Mr. John B. Kennedy for his inspiring address at our 44th Annual Banquet and thank Mr. Jack Knight for his enjoyable and educational talk.

We thank the exposition management and the advisory committee for their efforts in making this Exposition the largest and best in the history of the Society.

We thank the *American Society of Refrigerating Engineers* and the *National Warm Air Heating and Air Conditioning Association* for their cooperative effort in bringing our respective organizations, our members and our ideals in the art of heating, ventilating and air conditioning into more intimate contact.

Finally and with particular emphasis, we thank our good friend and Secretary, A. V. Hutchinson, and his very capable staff whose material effort underlies the execution of all work of the Society and whose fine spirit is so largely responsible for the familiar attitude of our members toward one another. The inadequacy of words alone makes it difficult to properly express the feeling in the hearts of the membership on whose behalf this resolution is gratefully submitted.

It was voted upon motion regularly made and seconded, that the Resolutions be adopted.

After calling for new business, to which there was no response, the presiding chairman, Mr. McIntire, adjourned the 44th Annual Meeting of the Society at 12:05 p. m.

The Exposition

The Fifth International Heating and Ventilating Exposition held at Grand Central Palace in New York, during the week of January 24-28, in conjunction with the 44th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, drew an attendance of over 44,300 registered guests who inspected the 327 exhibits of leading manufacturers. Among those attending were many members of the *American Society of Refrigerating Engineers* and the *National Warm Air Heating and Air Conditioning Association*, which held meetings in New York at that time.

The Exposition, which was held under the auspices of the A.S.H.V.E., was formally opened by D. S. Boyden, president of the Society and chairman of the Exposition Advisory Committee, on Monday, January 24.

With the prospect that 1938 is likely to bring about an increase in building, partly due to governmental encouragement to small home construction, thousands of builders, contractors, architects, engineers, dealers and distributors, and laymen were to be seen inspecting and comparing the various products in an effort to keep abreast of the times. Exhibitors reported unprecedented interest and actual sales in many cases.

In the oil burner field a variety of mechanical principles brought out the fact that oil burners are giving more heat and greater fuel economy than ever before.

The booths devoted to furnaces and boilers delighted visitors by the cascade of lively colors that greeted them as they walked down the aisles. In addition to their attractiveness, the boilers and furnaces offer greater economies in fuel con-

Some of the exhibits at the Fifth International Heating and Ventilating Exposition

sumption, more even distribution of heat, continuous hot water supply the year 'round, automatic operation, summer and winter air conditioning, quietness, safety, and easy servicing.

An innovation of note was the cooperative exhibit of solid fuel burners featuring the equipment of 32 different companies, almost half of whom had separate displays of their own. Effective, economical combustion of anthracite was thus given cooperative emphasis in terms of equipment adapted to varying types and sizes of installation.

Unit heaters and coolers, compact in design, easily installed, and in some instances portable, were featured in several exhibits.

In the field of summer air conditioning, displays ranged all the way from unitary coolers to central station cooling systems. Complete equipment was shown for cooling, dehumidifying, air circulating, air cleaning, humidifying, and heating.

Portable coolers for offices, shops, and individual rooms attracted wide interest. Several booths featured pumps, refrigerating equipment, water coolers and spray nozzles, fans, blowers, and exhausters.

Instruments and controls were displayed in number and most of them stressed the advantages of health, comfort, and ease of mind achievable by means of automatic control over temperature and humidity conditions.

The mechanical operation of valves, electrically operated pumps, pipes, fittings and tubing, grilles, louvres, registers, and filters, were displayed for inspection.

Insulating materials were featured and one exhibit included an actual size model of a roof, attic floor and outside wall in cross-section to show the insulation.

In the auxiliary fields of air conditioning, a variety of other equipment was shown including splash-proof motors, compressors, sheet metal tools, electric metal shears, corner-lock-forming machines, decorative metal parts for ranges and refrigerators, boiler cleaners, v-belts, electrical tools, ignition transformers, and a wide assortment of metals.

The Exposition was under the direction of Charles F. Roth, president, International Exposition Co., whose headquarters are at Grand Central Palace, New York.

New York Entertains

VISITING members, guests and ladies who attended the Society's 44th Annual Meeting can speak from experience and verify New York's reputation as a Wonder City, for during their short stay the Committee on Arrangements of New York Chapter had planned a program crowded with entertaining things to do.

On Monday at 9:30 a. m. registration began in the Fountain Court of the Hotel Biltmore where W. W. Timmis, chairman, and members of the Reception Committee extended cordial greetings.

Delegates from 25 Chapters met at a conference in the Cafe Moderne at 10:00 a. m., exchanged suggestions and plans, and discussed mutual problems. Pres. D. S. Boyden welcomed the delegates and introduced the following speakers: President-Elect E. Holt Gurney, R. C. Bolsinger, Thornton Lewis, and Prof. G. L. Larson.

At 2:00 p. m. the Officers and Council members, with the members of the Exposition Advisory Committee opened the 5th International Heating and Ventilating Exposition at Grand Central Palace.

The Bowman Room in The Biltmore was the scene of a Get-Acquainted Dinner at 6:30 p. m. Monday evening and 210 were present to be received by Pres. and Mrs. D. S. Boyden and to enjoy the music of Horace Heidt and his talented band and entertainers. The Entertainment Committee and G. E. Olsen, chairman, made their first formal appearance on this occasion.

Of the total registration, 97 ladies kept H. W. Fiedler and the other members of the Ladies Committee busy organizing a shopping tour and style show scheduled at 10:30 a. m., Tuesday, when 54 went to John Wanamaker's.

During the afternoon a suite of rooms on the first floor of the Biltmore hummed with the excitement of 10 tables of bridge. Tea was served and the holder of the highest score at each table received a clip pin sparkling with rhinestones.

A view of the gay White Way was enjoyed Tuesday night when the International Casino, one of New York's newest night clubs, was the scene of a combined party for members of the *National Warm Air Heating and Air Conditioning Association*, the A.S.H.V.E. and *American Society of Refrigerating Engineers*. The popularity of this event exceeded all expectations and 1,200 reservations were recorded. H. T. Richardson was master of ceremonies and with G. E. Olsen and D. L. Fiske handled all arrangements for this event.

After a late evening, or an early morning, 106 ladies left the hotel at 10:30 a. m. for a tour of Radio City and the N.B.C. Studios, where the wonders of this vast project including a broadcast were viewed and explained.

In the afternoon Father Malachy's Miracle, starring Al Shean, was enjoyed by 72 ladies at a theater party.

Wednesday evening the past presidents of the Society initiated Prof. G. L. Larson at a dinner in The Biltmore at 6:30 p. m. Those attending were Homer Addams, J. I. Lyle, W. H. Carrier, Thornton Lewis, J. D. Hoffman, J. F. Hale, W. H. Driscoll, W. T. Jones, F. B. Rowley, D. D. Kimball and H. P. Gant.

Not to be outdone by their illustrious husbands, Mrs. Homer Addams, Mrs. W. H. Carrier, Mrs. Thornton Lewis, Mrs. J. I. Lyle and Mrs. W. T. Jones had dinner together at the Advertising Club, where Mrs. D. S. Boyden was the hostess.

At 8:00 p. m. a lecture on Adventures in Science was given in the Electrical Association Auditorium at Grand Central Palace by Dr. Phillips Thomas, research engineer for Westinghouse Electric and Manufacturing Co. The filtering of air electrostatically, dry insulation, atom smashing, photo motor, breath relay and other educational and entertaining features were demonstrated.

On Thursday, January 27, a luncheon meeting of the Advisory Exposition Committee was held at the Chemists Club with C. F. Roth as host.

A busy day for the ladies began at 10:30 a. m. Thursday, when a party numbering 127 were taken on a special inspection tour of the Italian Liner S.S. Rex. This trip which was planned for the ladies also proved attractive to a number of the men.

From the pier the ladies were taken to Radio City Music Hall, where they were guests of the New York Chapter at a theater party. One hundred and fourteen were entertained by Snow White and the Seven Dwarfs, the first feature length Walt Disney production in techni-color.

The Inspection and Transportation Committee, of which A. J. Offner was chairman, had planned two trips for Thursday afternoon. The principal features of the construction and ventilating system of the Lincoln Vehicular Tunnel, connecting New York and New Jersey, were explained to 139 by A. C. Davis, Superintendent of Maintenance, Port of N. Y. Authority.

The air conditioning system of the Beechnut Packing Company Plant in Brooklyn, N. Y., was inspected by 46 and guides conducted the group through the factory.

The annual banquet on Thursday evening was held in the ballroom of The Biltmore. Activities began at 7:30 p. m. when the following, who were to be seated at the head table, started the procession: Mr. and Mrs. D. S. Boyden, Mr. and Mrs. E. Holt Gurney, A. J. Offner, John B. Kennedy, Jack Knight, W. H. Driscoll, Mr. and Mrs. Walter Heibel, Prof. J. D. Hoffman and A. V. Hutchinson.

W. H. Driscoll rendered capable service as toastmaster and kept the program in fast tempo. Prof. J. D. Hoffman, W. Lafayette, Ind., presented the past president's emblem to Col. D. S. Boyden, Boston, which he gratefully acknowledged. In presenting the emblem Professor Hoffman said:

This year's Annual Meeting impresses me with the Society's splendid growth in activities and in membership. I recall that in 1910, 28 years ago, when I had the privilege of being president, our membership was 372, which at that time was the largest of any previous year. Today it is nearly 3000.

The activities of the Society now parallel closely those in the commercial field. Increased activity in the industries stimulates our members to more earnest effort. Conversely, our educational and research work benefits the industries.

The growth of our membership roll tells the story of our growth in educational and research activity. Our meetings are now more largely attended, and the programs reflect more scientific thought and development than ever before. The research work, both in our Society's Research Laboratory and in the many cooperative projects with the technical universities and other groups, is outstanding for its careful development and its scientific advancement. Educational lectures in the field of heating and air conditioning have been added for the local groups requesting them. And not to carry this list too far, who can measure the ultimate value to the heating and air conditioning field of THE GUIDE? This work is an outstanding compendium of information for the whole field of heating and air conditioning.

It is evident that those who guide the destinies of such a Society under such conditions must be men of outstanding caliber and unselfish devotion.

Colonel Boyden, as president of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS during the past year, you have given largely of your time and energy and have guided the Society's interests through a most successful year. I congratulate the membership in having selected you one year ago as their standard bearer, and I now congratulate you upon a service cheerfully given and a work well done.

I feel myself honored at this time to have been selected as the spokesman of the Society to express to you the gratitude of its members for your services. I now present to you the Society's emblem of love and appreciation, which I know you will receive with satisfaction and cherish for the rest of your days.

The introduction of E. Holt Gurney, Toronto, brought forth an enthusiastic demonstration from a group of Canadian members in honor of the president-elect. Mr. Gurney spoke briefly and expressed his pleasure at being received and honored by those across the border in the United States.

The featured speaker of the evening was John B. Kennedy, N.B.C. commentator and advisory editor of *The Commentator*, whose voice is well known to radio listeners. Mr. Kennedy gave a philosophical talk on world conditions and was enthusiastically applauded by the audience.

Up in the Air was the subject of a talk by Jack Knight, ace pilot of United Air Lines, who reviewed the history and progress of commercial aviation during recent years.

The program was concluded with the presentation of jeweled pins by Mr. Driscoll to the following wives of members who were present: Mrs. Homer Addams, Germantown, Pa., whose husband had been a member of the Society for the longest time of anyone present; Mrs. H. G. Richardson, Salt Lake City, Utah, for coming the greatest distance in the United States; Mrs. T. L. Livingston, Hamilton, Ont., for coming the greatest distance in Canada; Mrs. J. H. Van Alsburg, Chicago, Ill., as the most recently married.

Music by Phil Sands and his orchestra was enjoyed during dinner and later for dancing. Russell Donnelly, chairman, and members of the Banquet Committee successfully carried out their difficult assignment and arranged for banquet service to 500 members, ladies and guests.

The Finance Committee, of which F. E. W. Beebe was chairman, and the Publicity Committee, C. H. B. Hotchkiss, chairman, were on the job every day.

A. C. Buensod, general chairman, and Alfred Engle, vice-chairman, of the Committee on Arrangements were on duty throughout the meeting to see that all plans were effectively carried through.

On Friday after adjournment of the meeting at noon, the Council met for luncheon in the Cafe Moderne, and many members spent the afternoon and evening at the 5th International Heating and Ventilating Exposition.

On Sunday, January 23, Pres. D. S. Boyden was guest speaker on the Armco program at 3:00 p. m. over a national network through station WJZ. Newly elected Pres. E. Holt Gurney, W. L. Fleisher, chairman of the Committee on Research, and A. C. Davis were featured on a broadcast over the Mutual System through WOR at 12:45 p. m., Friday, January 28.

As a fitting climax to the 1938 Annual Meeting a cruise party for a six-day trip to Bermuda sailed from New York, January 29, at 3:00 p. m. on the Monarch of Bermuda. Among the members of the A.S.H.V.E. group who went to Bermuda were Mr. and Mrs. N. D. Adams, Mr. and Mrs. A. L. Browne and daughter, Mr. and Mrs. W. S. Eisele, Mr. and Mrs. A. E. Stacey, Jr., Mr. and Mrs. J. H. Walker and Messrs. P. H. Foster, L. J. Fowler, W. G. Case and William Glass.

The New York Chapter enjoyed the opportunity of entertaining fellow members, guests and ladies from other parts of the United States, Canada, England, Germany, French Indo-China and Denmark, and hopes that they may long remember their visit to New York with a great deal of pleasure.

Correlating Thermal Research

As a part of the efforts of the A. S. H. V. E. Research Laboratory to correlate research in thermal engineering carried on by the many institutions engaged in such work, and to disseminate the published results of such studies together with other reports of progress in the field, and in order to make this information available to the membership of the Society, there will be published monthly on this page a limited number of brief abstracts of articles which it is believed will be of interest to all concerned. For more complete lists address the Librarian, A. S. H. V. E. Research Laboratory, U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.

F. C. Houghten, *Director*

W. L. Fleisher, *Chairman,*
COMMITTEE ON RESEARCH.

• Building Code Correlating Committee Shows Development of Building Projects, by Rudolph P. Miller. *Industrial Standardization,* Vol. 8, No. 12, Dec. 1937, pp. 331-332. Discussion of status of eight projects authorized under the supervision of the Building Code Correlating Committee, as follows:—Recommendations for masonry, A51-Requirements for fire protection and fire resistance, A52-Requirements for chimneys and heating appliances, A53-Requirements for light and ventilation, A54-Requirements for fire extinguishing equipment, A55-Administrative requirements for building codes, A56-Requirements for excavations and foundations, A57-Building code requirements for iron and steel. Advisory committee working on stresses.

• Thermal Conductivity of Glass, Chilled Glass Quartz, Fused Quartz (transparent), Coal, Isolite, Proselain etc., by Shiro Nukiyama. *Transactions Society of Mechanical Engineers,* Japan. Vol. 2, No. 8, pp. 344-345, 1936. Abstracted in *Physik.,* Ber., Vol. 18, No. 14, p. 1286, 1937. The measurement of heat conductivity by a relative comparison method is described. The same heat flows successively through the comparison plate with known heat conductivity and the plate to be determined in the range of 0 to 100 C.

• Special Glass in the Lighting Field, by R. Schmidt. *Glastech.,* Ber., Vol. 15, No. 3, pp. 98-99, 1937. A report of the development and present status of special glasses transmitting and absorbing ultra-violet light. Fluorescent glasses, metal vapor coated glasses and high softening temperature glasses are included. Tables of characteristic glass compositions are given. Also a bibliography.

• Spun Glass for Insulation of Railway Equipment, by Nordmann. *Chem. Fabrik,* Vol. 10, pp. 299-300, 1937. See also *Chem. Abstracts,* Vol. 31, p. 6762, 1937. The method of applying the insulation to locomotives and steam lines is described. Comparative tests on passenger locomotives, insulated by the usual air space and insulated with 50 milli-meters of mineral wool, at 60 to 80 kilo-meters/hour showed air insulation reduced loss 50 per cent and mineral wool saved 40 to 43 per cent more than the air. Sufficient data not yet available.

• Italy; Land of Prototypes for Glass Application, by G. Schieb. *Oesterr. Glaserztg.,* Vol. 2, pp. 101-104, 1937. Architectural applications, including all-glass doors, are illustrated.

• Hospital Air Conditioning, by C. P. Yaglou. *The Journal of Industrial Hygiene and Toxicology.* Vol. 18, No. 10, Dec. 1936, pp. 741-766, 8 figs. Bibliography. Air conditioning is now applied to four or five wards in modern hospital where it has been shown to be of sufficient value to justify its use. Discussion of air conditioning in nurseries for premature infants, anesthesia and operating rooms, oxygen therapy chambers, heat therapy chambers or cabinets, and allergic wards.

• Water Hammer Symposium. *Mechanical Engineering,* Vol. 59, No. 11, Nov. 1937, p. 871. Program of *A.S.M.E.* Annual Meeting for Dec. 1937, for list of papers presented. See also *Power,* Mid-Dec. 1937, Vol. 81, No. 14, pp. 812-3 for discussion of meeting. 16 papers and reports by an international group of specialists discuss water-hammer problems of water power, pumps, water-conduits and similar problems of liquid flow in diesel fuel-injection systems.

• Measurement of Radiant Energy, edited by W. E. Forsythe, many contributors. Published by the McGraw-Hill Book Company, Inc., New York and London, 1937, 452 pages. Prepared under direction of Committee on Methods of Measurement of Radiation of the Division of Physical Sciences, National Research Council. Main discussion on radiation of and methods of measuring radiation from incandescent lamps. Short discussion on solar radiation.

• Symposium on Waste Heat Boilers. *The Institute of Fuel,* Vol. XI, No. 56, Dec. 1937, pp. 81-101. Many illustrations. Waste Heat Boilers in the Steel Industry, by A. F. Webber. Waste Heat Boilers, by Major W. Gregson. Waste Heat Recovery using Thimble-tube and Water-tube Boilers, by E. F. Spanner. Waste-heat Recovery in Gas-fired Practice, by James W. Reber.

• Air Currents and Drafts as Factors in Air Conditioning, by D. L. MacLean and Ruth C. Partridge. *The Journal of Industrial Hygiene and Toxicology,* Vol. 19, No. 10, Dec. 1937, pp. 562-570. Report of study of air movements. Threshold air movement or least air movement; uncomfortable air movement or draft. Also physiological effects of draft and changes of nose membrane when face is exposed to draft.

• Bibliography of U. S. Bureau of Mines Investigations on Coal and Its Products, 1910-1935, by A. C. Fieldner, Alden H. Emery and M. W. von Bernewitz. U. S. Bureau of Mines Technical Paper No. 576, 145 pages. Price 5c from Supt. of Documents, Government Printing Office, Washington, D. C.

• Thermal Insulating Material from Vermiculite, by N. Subarov. *Novosti Tekniki* (U.S.S.R.), No. 14, pp. 39-40, 1937. Abstracted in Referat. Silikatliteratur. Vol. 4, No. 8, 1937, p. 4327. Vermiculite expands 20 times its original volume on heating because of the evaporation of its water content. Rich deposits of vermiculite in U.S.S.R. in the Ural Mts. It has mechanical strength lower than cork products, but is more hygroscopic, heat conductivity (0.060) is higher than cork (0.045).

• Symposium of Ignition of Fuel on Grates. *The Institution of Fuel,* Vol. XI, No. 56, Dec. 1937, pp. 102-133, many illustrations. The Ignition of Coal on a Grate, by P. O. Rosin and H. R. Fehling. Contribution to the Study of the Ignition of Fuel Beds, by A. C. Dunningham and E. S. Grumell. The Ignition of Coal, by Clarence A. Seyler and Thomas E. Jenkins. Mathematical discussions.

• Nomograph for Ventilation Problems, by A. S. Ford. *Colliery Engineering,* Vol. 15, No. 167, Jan. 1938, pp. 4-5. Nomograph designed for convenient solution of ventilation problems commonly encountered in mining practice.

• An Investigation of the Bacterial Contamination of the Air of Textile Mills with Special Reference to the Influence of Artificial Humidification, by W. F. Wells and E. C. Riley. *The Journal of Industrial Hygiene and Toxicology,* Vol. 19, No. 10, Dec. 1937, pp. 513-561. Bibliography. Study and measurement of bacterial contamination in textile mills.

• Air Conditioning Problems in the Citrus Fruit Industry, by Herbert Bishop Addington. *Heating and Ventilating,* Vol. 34, No. 12, Dec. 1937, pp. 53-56. Description of air conditioning to prepare citrus fruit for the West Coast market. Methods used with data on time and heat quantities required. Diagram of packing, cooling and coloring operations.

NEWS OF LOCAL CHAPTERS

OFFICERS OF LOCAL CHAPTERS—1938

ATLANTA: *Organized,* 1937. *Headquarters,* Atlanta, Ga. *Meets,* First Tuesday. *President,* E. W. KLEIN, 152 Nassau St., N. W. *Secretary,* C. T. BAKER, 713 Glenn St., S. W.

CINCINNATI: *Organized,* 1932. *Headquarters,* Cincinnati, O. *Meets,* Second Tuesday in Month. *President,* I. B. HELBURN, 610 Chamber of Commerce Bldg. *Secretary,* H. E. SPROULL, 1005 American Bldg.

GOLDEN GATE: *Organized,* 1937. *Headquarters,* San Francisco, Calif. *Meets,* First Tuesday. *President,* B. M. WOODS, Univ. of Calif., Berkeley, Calif. *Secretary,* G. J. CUMMINGS, 113 Tenth St., Oakland, Calif.

ILLINOIS: *Organized* 1906. *Headquarters,* Chicago, Ill. *Meets,* Second Monday. *President,* S. I. ROTTMAYER, 407 S. Dearborn St. *Secretary,* C. E. PRICE, 6 N. Michigan Ave.

IOWA-NEBRASKA: *Organized,* 1937. *Headquarters,* Omaha, Neb. *President,* M. J. STEVENSON, 1643 South 20th St., Lincoln, Neb. *Secretary,* W. R. WHITE, 4339 Larimore Ave., Omaha, Neb.

KANSAS CITY: *Organized,* 1917. *Headquarters,* Kansas City, Mo. *Meets,* Second Monday in Month. *President,* A. H. SLUSS, 827 Mississippi Ave., Lawrence, Kan. *Secretary,* GUSTAV NOTTBERG, 914 Campbell St.

MANITOBA: *Organized,* 1935. *Headquarters,* Winnipeg, Man. *Meets,* Fourth Thursday. *President,* D. F. MICHIE, 492 Wardlaw Ave. *Secretary,* E. J. ARGUE, Ste. 23, Estelle Apts.

MASSACHUSETTS: *Organized,* 1912. *Headquarters,* Boston, Mass. *Meets,* Third Tuesday in Month. *President,* JAMES HOLT, Massachusetts Institute of Technology, Cambridge, Mass. *Secretary,* H. C. MOORE, 69 Massachusetts Ave., Cambridge, Mass.

MICHIGAN: *Organized,* 1916. *Headquarters,* Detroit, Mich *Meets,* First Monday after the 10th of the Month. *President,* F. J. FEELY, 950 Trombley Rd., Grosse Pointe Pk. *Secretary,* G. H. TUTTLE, 2000 Second Ave.

WESTERN MICHIGAN: *Organized,* 1931. *Headquarters,* Grand Rapids, Mich. *Meets,* Second Monday in Month. *President,* W. W. BRADFIELD, 901 Michigan Trust Bldg. *Secretary,* S. W. TODD, JR., 309 Paris, S. E.

MINNESOTA: *Organized,* 1918. *Headquarters,* Minneapolis, Minn. *Meets,* Second Monday in Month. *President,* R. E. BACKSTROM, Room 1981, First Natl. Bank Bldg., St. Paul, Minn. *Secretary,* F. C. WINTERER, 836 Juno St., St. Paul, Minn.

MONTREAL. *Organized,* 1936. *Headquarters,* Montreal, Que. *Meets,* Third Monday. *President,* G. L. WIGGS, University Tower. *Secretary,* C. W. JOHNSON, 630 Dorchester St., W.

NEW YORK: *Organized,* 1911. *Headquarters,* New York, N. Y. *Meets,* Third Monday in Month. *President,* W. E. HEIBEL, 11 West 42nd St. *Secretary,* T. W. REYNOLDS, 100 Pinecrest Dr., Hastings-on-Hudson, N. Y.

WESTERN NEW YORK: *Organized,* 1919. *Headquarters,* Buffalo, N. Y. *Meets,* Second Monday in Month. *President,* B. C. CANDEE, 19 Tremont Ave., Kenmore, N. Y. *Secretary,* W. R. HEATH, 119 Wingate Ave.

NORTHERN OHIO: *Organized,* 1916. *Headquarters,* Cleveland, O. *Meets,* Second Thursday in Month. *President,* PHILIP COHEN, 401 East Ohio Gas Bldg. *Secretary,* C. A. MCKEEMAN, Case School of Applied Science.

OKLAHOMA: *Organized,* 1935. *Headquarters,* Oklahoma City, Okla. *Meets,* Second Monday. *President,* E. F. DAWSON, University of Oklahoma, Norman, Okla. *Secretary,* E. W. GRAY, Box 1498, Oklahoma City, Okla.

ONTARIO: *Organized,* 1922. *Headquarters,* Toronto, Ont. *Meets,* First Monday in Month. *President,* G. A. PLAYFAIR, 113 Simcoe St. *Secretary,* H. R. ROTH, 57 Bloor St. W.

PACIFIC NORTHWEST: *Organized,* 1928. *Headquarters,* Seattle, Wash. *Meets,* Second Tuesday in Month. *President,* W. W. COX, 326 Columbia St. *Secretary,* M. N. MUSGRAVE, 314-9th Ave., N.

PHILADELPHIA: *Organized,* 1916. *Headquarters,* Philadelphia, Pa. *Meets,* Second Thursday in Month. *President,* L. P. HYNES, 240 Cherry St. *Secretary,* C. B. EASTMAN, 530 Brookview Lane, Brookline, Upper Darby, Pa.

PITTSBURGH: *Organized,* 1919. *Headquarters,* Pittsburgh, Pa. *Meets,* Second Monday in Month. *President,* J. F. COLLINS, JR., 435 Sixth Ave. *Secretary,* T. F. ROCKWELL, Carnegie Inst. Tech.

ST. LOUIS: *Organized,* 1918. *Headquarters,* St. Louis, Mo. *Meets,* First Tuesday in Month. *President,* G. W. F. MYERS, 3947 W. Pine Blvd. *Secretary,* D. J. FAGIN, 1344 Woodruff Ave.

SOUTHERN CALIFORNIA: *Organized,* 1930. *Headquarters,* Los Angeles, Calif. *Meets,* Second Tuesday in Month. *President,* E. H. KENDALL, 1978 S. Los Angeles St. *Secretary,* J. F. PARK, 1234 South Grand.

TEXAS: *Organized,* 1936. *Headquarters,* College Station, Texas. *President,* R. F. TAYLOR, 909 Banker's Mortgage Bldg., Houston, Tex. *Secretary,* W. H. BADGETT, Texas Engrg. Experiment Station, College Station, Tex.

WASHINGTON, D. C.: *Organized,* 1935. *Headquarters,* Washington, D. C. *Meets,* Second Wednesday in Month. *President,* L. OURUSOFF, 411 Tenth St., N. W. *Secretary,* L. F. NORDINE, Room 203, 734 Jackson Pl., N. W.

WISCONSIN: *Organized,* 1922. *Headquarters,* Milwaukee, Wis. *Meets,* Third Monday in Month. *President,* J. H. VOLK, 1906 W. St. Paul Ave. *Secretary,* H. C. FRENTZEL, 3000 W. Montana St.

C.-E. A. Winslow Addresses Buffalo Meeting

January 11, 1938. Western New York Chapter met at the University Club and 44 members and guests were present for dinner which preceded the meeting.

Minutes of the November and December meetings were read and approved and the proposed amendment presented at the November meeting was adopted and upon approval of the Society's Council will become a part of the Chapter's Constitution.

M. C. Beman was called upon and introduced the speaker, Professor Winslow, director of the John B. Pierce Laboratory of Hygiene at Yale University. The subject of Dr. Winslow's talk was Physiological Reactions and Sensations of Pleasantness under Varying Atmospheric Conditions, and this was well received by members and guests who had a number of questions in the discussion which followed.

Southern California Chapter Holds Joint Meetings with Local A.S.R.E. Group

January 18, 1938. One hundred members and guests of the A.S.H.V.E. and *A.S.R.E.* met in joint session at Calwell's Restaurant, Los Angeles, and Vice-Pres. H. M. Hendrickson of the A.S.H.V.E. presided in the absence of E. H. Kendall who was in the east to attend the 44th Annual Meeting of the Society.

Following the reading of the minutes and routine business, Mr. Hendrickson introduced Herman Seid, who spoke on the subject of Patented Air Conditioning Systems. In addition to discussing the subject, Mr. Seid gave some idea of the trend of regulations being proposed in the east, especially in Washington, D. C. and called upon the industry to protect itself from excessive regulation by maintaining high standards of design.

Secy. J. F. Park reports that the meeting adjourned at 9:30 p.m. after some discussion by those in attendance.

December 14, 1937. The regular meeting of the Southern California Chapter was held at Calwell's Restaurant with President E. H. Kendall presiding, and was held jointly with the local section of the *A.S.R.E.*

The minutes of the previous meeting were read and approved and communications regarding the reinstatement of initiation fees by the A.S.H.V.E. were read.

President Kendall was elected delegate to the Society's Annual Meeting in New York, January 24-28.

The Chapter was honored by the presence of Prof. E. F. Dawson, University of Oklahoma, who spoke on the subject of Refrigeration, Refrigerants and Air Conditioning. Professor Dawson's address was well received and gave an indication of future interesting programs which might be furnished by the Speakers' Bureau.

Following the address there was a discussion by members and guests who made up the attendance of 95.

Coal Subject of Montreal Chapter Meeting

January 17, 1938. The regular meeting of Montreal Chapter was held at the Windsor Hotel with 50 members and guests attending. Pres. G. L. Wiggs called the assembly to order and introduced the guests who were present.

F. A. Combe, consulting engineer, opened his talk with a history of coal and its formation, explaining its effect on the present coal classification. After giving a description of the fuels available for combustion work, he showed their characteristics as applied to a domestic furnace, illustrating his talk with slides. Mr. Combe then discussed various types of modern boilers and their construction. He concluded with the recommendation that all boilers be rated on a steam quantity basis rather than by boiler horsepower.

After the lecture an open discussion was held and a vote of thanks was given on behalf of the chapter by T. H. Worthington.

Minutes of the previous meeting were read and a vote was taken on the amendment previously presented that all new chapter members be allowed a discount of $1.00 on chapter dues, if payment is made within one month after billing, and this motion was carried.

L. H. Laffoley made an announcement regarding the Annual Meeting of the Society in New York and requested that every member who could possibly do so should be present.

The meeting adjourned at 10:15 p.m. according to the report of Secy. C. W. Johnson.

Philadelphia Chapter Enjoys Social Meeting

January 13, 1938. The meeting of the Philadelphia Chapter was held at Bookbinder's Restaurant and the reading of the minutes and the regular business procedure were dispensed with.

During the dinner Pres. L. P. Hynes introduced A. V. Hutchinson, secretary of the A.S.H.V.E., who was present by invitation and Mr. Hutchinson acknowledged his pleasure at being with Philadelphia members and asked R. C. Bolsinger to tell about the program and the Society's 44th Annual Meeting in New York, January 24-28.

Following the dinner Harry Erickson, chairman of the Meetings Committee, presented a program consisting of several acts by professional entertainers.

Secy. C. B. Eastman reports that the meeting broke up at an early hour in a spirit which indicated that a good time was had by all.

English Heating and Ventilating Practice Discussed by Michigan Chapter

January 17, 1938. There were 125 members and guests present for dinner and a meeting of Michigan Chapter held in the general offices of the Detroit Edison Co., Detroit. Pres. F. J. Feely called the meeting to order and asked for reports of the secretary, treasurer and chairman of the Program and Membership Committees. E. E. Dubry outlined meetings for the remainder of the season and W. G. Boales introduced all new members who had affiliated with the A.S.H.V.E. during the past year.

Mr. Dubry then introduced the speaker of the evening, Douglas Dow. In discussing the subject, Some Observations of English Heating and Ventilating Practice, Mr. Dow pointed out that English people are accustomed to lower indoor temperatures during the heating period than is usual in the United States and that this is caused by the fact that humidities are considerably higher, the people are more active physically, and generally speaking the homes are believed to be of better construction. He also described some of the installations of radiant heating being made at the present time.

After the talk and discussion, Secy. G. H. Tuttle reports that a movie was shown before the meeting adjourned at 10:30 p.m.

John Howatt Visits Iowa-Nebraska Chapter

January 11, 1938. At a meeting held at the Rome Hotel, Omaha, when 35 members and guests were present, Iowa-Nebraska Chapter was honored by having John Howatt, Chicago, Ill., present as guest and speaker. In the absence of Pres. M. J. Stevenson, Vice-Pres. A. L. Walters presided.

Among monthly matters and routine business was a letter from Prof. R. A. Norman, Iowa State College, regarding the participation of the Iowa-Nebraska Chapter in the annual spring heating, ventilating and air conditioning conference. Mr. Walters stated that a meeting at the Iowa State College at that time was being considered and there was some discussion regarding the activities of different organizations and industries.

F. L. D. Banner said that the Omaha Engineers Club offered its cooperation to the A.S.H.V.E. Iowa-Nebraska Chapter and Mr. Walters appointed Mr. Banner to investigate this matter.

Mr. Walters then introduced the speaker of the evening, John Howatt, past president of the A.S.H.V.E. and chief engineer of the Board of Education, City of Chicago.

Mr. Howatt presented a very interesting discussion on the subject, A Few Air Conditioning Engineering Problems and Answers. The opinions that he gave on this broad topic were both interesting and educational. Some of the facts he mentioned regarding advance in engineering research were surprising, especially his statement that the Golden Gate Bridge, if made of the same material used in the George Washington Bridge, would not stand up under its own weight. His engineering philosophy and opinions regarding questions of national scope were most interesting, and will be long remembered by many of those who attended. It was considered a privilege to have the speaker present.

At the conclusion of Mr. Howatt's talk, Mr. Walters expressed the appreciation of the Iowa-Nebraska Chapter for his coming to Omaha, and for the splendid talk that he had made.

According to the report of Secy. W. R. White the meeting adjourned at 10:15 p.m.

Cincinnati Chapter Hears Talk on Air Conditioning

January 11, 1938. Seventy-five members and guests attended the regular meeting of the Cincinnati Chapter at the Engineers Club. Dinner was served preceding the meeting at 6:30 p. m. with 24 members present at the Gibson Hotel rathskeller.

Vice Pres. O. W. Motz presided in the absence of Pres. I. B. Helburn. After the business of the evening was dispensed with, Mr. Motz introduced the speaker of the evening, Herman Seid, Auditorium Conditioning Corp., who gave an interesting address on the subject of Air Conditioning.

Although Mr. Seid had prepared a written paper, which was distributed to the members, he talked extemporaneously with the aid of diagrams on a blackboard.

He pointed out that what people want from air conditioning is comfort, and not temporary relief, so that in designing a system, careful consideration should be given to ventilation, refrigeration, control of the equipment, and good distribution of the air without drafts. Mr. Seid explained several systems of

by-pass, and gave examples of installations. He also predicted that absorption methods using silica gel, lithium chloride or activated alumina were on the increase. A lively discussion followed Mr. Seid's talk.

Mr. Motz then introduced Charles Lawrence, also of the Auditorium Conditioning Corp. of New York, who gave a few words of greeting.

According to Secy. H. E. Sproull's report the meeting adjourned at 10:30.

Speakers at Pittsburgh Meeting Discuss Draft and Heat Flow Studies

January 10, 1938. Pres. J. F. Collins, Jr. called the meeting of the Pittsburgh Chapter to order at 7:45 p. m. in the private dining room of Stouffer's Restaurant.

After the secretary's report which was read and approved, J. E. McLean read a statement showing the Chapter's financial condition, following which E. C. Smyers reported for the Program Committee.

Before proceeding with the technical program, M. L. Carr, chairman of the A.S.H.V.E. Technical Advisory Committee on Air Conditioning Requirements of Glass, read a report covering the committee's personnel, its objectives and its accomplishments.

F. C. Houghten then presented an abstract of the laboratory paper, which was given before the Annual Meeting of the Society held in New York, in connection with the studies made on the use of skin temperature measurements to determine what constitutes a draft.

Leighton Orr, research engineer for the Pittsburgh Glass Co., was the second speaker and his subject dealt with results of an investigation made on heat flow through different types of steel frame windows including single and double glazing.

Thirty-five members and guests participated in the general discussion which followed these two talks and the meeting adjourned at 9:50 p. m. according to the report of Secy. T. F. Rockwell.

Illinois Chapter Discusses Panel Heating

January 10, 1938. Following dinner at the Brevoort Hotel Pres. S. I. Rottmayer presided at the regular meeting of the Illinois Chapter at which 135 were present.

After the reading of the minutes Tom Brown reported for the Meetings and Publications Committee and E. M. Mittendorff, chairman of the Membership Committee, introduced two new members—T. J. Killian and R. C. White.

T. N. Adlam, chief engineer, Sarco Co., Inc., New York, was the speaker of the evening and addressed the Illinois Chapter on the subject of Panel Heating. Also appearing on this program was Irving Brooke, member of the Illinois Chapter and consulting engineer, who was accompanied by Denison B. Hull, architect and owner of the only home in the Chicago district with panel heating.

Mr. Adlam discussed the theory of radiant or panel heating, which aims to get a mean surface temperature in walls and ceilings to regulate the flow of heat from the body. He showed a number of slides which pictured the equipment used, the methods of installation, the calculation of surface requirements and many examples of panel heating in a wide variety of buildings.

Mr. Brooke then gave an account of his experiences with Mr. Hull's house and Mr. Hull testified to the satisfaction he was enjoying with the panel heating system.

A lively discussion followed this interesting talk and the meeting adjourned at 10.30 p. m. according to the report of Secy. C. E. Price.

Fairbanks Morse Moves General Offices

The general offices of Fairbanks Morse & Co. are now located at 600 S. Michigan Ave., Chicago, where they moved earlier this year. The first five floors of the modernized building are occupied by the executive, sales and departmental offices, and a display room is located on the first floor.

G. E. Air Conditioning Institute

D. W. McLenegan, assistant commercial engineer, air conditioning department, General Electric Co., has been named manager of the company's Air Conditioning Institute, and will be assisted by R. D. Wood, according to an announcement by J. J. Donovan. Distributors, dealers, and others closely associated with the design, manufacture and installation of G-E equipment will take part in the institute's activities. In the quarters which have been constructed on the roof of the present main building of the plant at Bloomfield, N. J., an auditorium, offices and an exhibit room are included, and a complete line of heating and air conditioning equipment will be on display.

Oil Burner Institute Appoints President

F. H. Dewey, general manager, air conditioning division, Gar Wood Industries, Inc., Detroit, was appointed president of the *Oil Burner Institute* at a special meeting of the board of directors, to fill the unexpired term of C. M. Lockwood, Rock Island, Ill., who resigned, according to G. H. Porter, managing director of the *O. B. I.*

S. A. Jellett, Jr. Announces New Business Connection

Stewart A. Jellett, Jr. has announced his connection with the firm Baizley & Jellett as manufacturers representatives at 727 Penfield Building, Philadelphia, Pa. The business of this firm is intended to be confined principally to the distribution of heating, ventilating and air conditioning apparatus in the vicinity of Philadelphia.

A.S.H.V.E. Member Returns to French Indo China

P. E. Cox sailed for England on the Queen Mary on his way to French Indo China. He anticipates visiting London, Paris and Amsterdam, and then proceeding by aeroplane from Paris to French Indo China.

He attended the Society's 44th Annual Meeting and travelled 14,000 miles via Hong Kong, Shanghai, Kobe, Yokohama, Vancouver and Seattle.

Johnson Service Co. President Retires

H. W. Ellis, Milwaukee, Wis., retired on February 1st from his position as president and general manager of the Johnson Service Company, manufacturers and contractors for automatic temperature control systems. This marks the termination of 43 years of service in the Johnson organization, the last 26 of which were spent as the active directing head of the business. Mr. Ellis assumes the newly created position of chairman of the Board of Directors. In 1936 he was elected a Life Member of the A. S. H. V. E., which he joined in 1923.

J. A. Cutler, Chicago, succeeds Mr. Ellis as president and general manager at Milwaukee. Mr. Cutler became a sales engineer for the Johnson Company in 1912, advancing to the position of manager of the central district and of the Chicago office in 1922. He continued to occupy that position until his recent advancement, although being given additional duties as vice-president several years ago and as general sales manager in 1936. O. G. Ward, manager of the Denver office, is to take Mr. Cutler's place as vice-president and district manager at Chicago. He, like Mr. Cutler, has been in the service of the com-

pany for 25 years, having been manager of the Cleveland office before going to Denver. A. W. Cooper, from the Los Angeles office, replaces Mr. Ward as manager at Denver.

The retirement of Kennedy Duff, after many years as eastern manager, has resulted in the appointment of M. F. Rather as manager of the Eastern District at New York. Mr. Rather began his service as a sales engineer in 1916 and has been manager of the Cleveland office for more than 15 years. He has been replaced as manager at Cleveland by P. D. Gayman, who has been manager at Indianapolis for several years. H. V. Alexander, sales engineer from the Buffalo office, becomes manager at Indianapolis.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible..

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 29 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by March 15, 1938, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
BAKER, W. C., Air Cond. Engr., Frigidaire Div., Nashville, Tenn.	Albert Buenger C. E. Lewis	W. A. Chapman, Jr. Rothwell Woodward
CALL, JOSEPH, Air Cond. Engr., Elliott-Lewis Co., Philadelphia, Pa. (*Advancement*)	M. G. Kershaw R. E. Payne	R. J. Steel A. Chance (*Non-Member*)
CARON, HECTOR, Engr., Rochelle, Ill.	R. W. Menk (*Non-Member*) C. Schaefer (*Non-Member*)	J. J. Humove (*Non-Member*) W. E. White (*Non-Member*)
DAVIS, CHARLES, Chief Engr., Rathe Htg. Corp., New York, N. Y.	E. J. Ritchie	Albert Goldhammer (*Non-Member*)
	Abraham Abrams	H. E. Rathe (*Non-Member*)
GOSSETT, A. L., Dealer, Sales Promotion, Southwest Airtemp Corp., Oklahoma City, Okla.	H. A. Hunter (*A. S. C. E.*) Earl Glasgow (*A. S. C. E.*)	J. C. Harrison (*A. S. R. E.*) D. D. Price (*Non-Member*)
HAGAN, W. V., Secy. & Mgr., V. J. Hagan Co., Sioux City, Iowa (*Advancement*)	W. R. White A. L. Walters	Henry Kleinkauf R. A. Norman
KLEINKAUF, HENRY, Br. Mgr., Natkin & Co., Omaha, Nebr. (*Advancement*)	Sidney Pines M. A. Disney	L. M. Farber Henry Gould
KRUSE, W. G., JR., Owner, Kruse Engrg. Co., Newark, N. J.	Roswell Farnham J. H. Hucker	H. C. Murphy J. H. Milliken
LISKOW, J. G., Chief Engr., Claude B. Schneible Co., Chicago, Ill.	J. F. McIntire A. P. Darlington	R. F. Connell W. A. Russell
MCINTIRE, J. L., Sales Engr., York Ice Machinery Corp., St. Louis, Mo.	L. J. Du Bois R. R. Irwin	A. G. Matousek E. F. Weber
MCLAREN, T. H., Gen. Sales Mgr., James Morrison Brass Mfg. Co., Ltd., Toronto, Ont., Can.	S. W. Alexander Fred Ellis	E. R. Gauley J. H. Fox
MILLER, J. E., Sales Engr., Fairbanks Morse Co., St. Louis, Mo.	J. H. Van Alsburg	J. W. Bostwick (*Non-Member*)
	C. H. Dean	W. B. Kella (*Non-Member*)
MITCHELL, J. A., Dist. Repr., Minneapolis-Honeywell Reg. Co., Austin, Minn.	L. C. Gross M. L. Todd	J. C. Knox N. D. Adams
MORSE, L. S., JR., Sales Engr., Westerlin & Campbell Co., Detroit, Mich. (*Advancement*)	H. E. Paetz J. R. Hertzler	E. L. Hogan W. E. Barnum, Jr.
NOLAN, R. E., Owner, Ralph Nolan, Atlanta, Ga.	W. J. McKinney H. G. Krayenhof	E. W. Klein R. F. Hahn
NORRIS, W. P., Sales Engr., Natkin & Co., St. Louis, Mo.	Benjamin Natkin L. W. Moon	Sidney Pines S. J. Shure (*A. S. R. E.*)
OOSTEN, L. S., Draftsman, Bell & Gossett Co., Chicago, Ill.	E. J. Gossett R. E. Moore	H. A. Lockhart C. E. Pullman (*Non-Member*)
PARVIS, R. S., Engr., Diamond Ice & Coal Co., Wilmington, Dela.	G. P. Weatherlow B. F. Lownsbery	R. C. Bolsinger A. H. MacDade

CANDIDATES

PLANT, E. B., Asst. Engr., Can. Pacific Ry. Co., Montreal, Que., Can.

QUICK, B. A., Sales Mgr., The Independent Register Co., Cleveland, Ohio

REIFSCHNEIDER, JAKE, Supt. of Maintenance, Eppley Hotels Co., Omaha, Nebr.

RUFF, H. A., Pres. & Treas., Herbert A. Ruff, Inc., Buffalo, N. Y.

SAUER, E. O., Htg. & Air Cond. Engr., Detroit Inst. of Tech., Detroit, Mich.

STROUSE, S. W., Sales Mgr., Cooney Refrigeration Co., Buffalo, N. Y. (*Reinstatement*)

THOMAS, R. C., Vice-Pres. & Mgr., Bonair Co., Inc., West Norfolk, Va.

UPSON, W. L., Director of Research, Torrington Mfg. Co., Torrington, Conn.

VOLKHARDT, A. N., Br. Mgr., Alfred L. Hart, Inc., Stapleton, S. I., N. Y.

YARBOROUGH, T. R., Dist. Sales Engr., Frigidaire Div., Gen. Motors Sales Co., Atlanta, Ga.

YEAZEL, G. A., Air Cond. Engr., Sampson Electric Co., Chicago, Ill.

REFERENCES

Proposers	*Seconders*
G. L. Wiggs L. H. Laffoley	F. G. Phipps W. W. Timmins
J. E. Maynard F. R. Bishop	A. P. Kratz S. Konzo
Henry Kleinkauf Maurice Peiser	W. R. White H. W. Hurley (*Non-Member*)
C. H. Lighthart W. S. Eisele	B. C. Candee J. J. Landers
Matthew Soeters Alger Chapman (*Non-Member*)	C. C. Winn (*S. P. E. E.*) Luke Broughton (*Non-Member*)
D. J. Mahoney M. C. Beman	B. C. Candee P. S. Hedley
T. J. Atkins A. K. Scribner (*A. C. S.*)	A. H. Eustis (*A. S. R. E.*) W. M. Lofton, Jr. (*A. C. S.*)
Alexis Doster W. G. Boales	W. L. Fleisher W. A. Danielson
Bernard Leventhal Irwin Jalonack	O. K. Wahrenbrock W. T. Wrightson
W. J. McKinney H. K. McCain	H. G. Krayenhof R. F. Hahn
Leonard Kemph (*Non-Member*)	Gilbert Stadeker (*Non-Member*)
Howard Somers (*Non-Member*)	Benjamin Greenhouse (*Non-Member*)

Candidates Elected

In the past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

ABBOTT, T. J., Vice-Pres., George C. Abbott, Ltd., Toronto, Ont., Can.

BECKER, R. K., Dept. Mgr., Ohio Valley Hdwe. & Roofing Co., Evansville, Ind.

BENSON, M. L., Mgr. Air Cond. Coil Div., McQuay, Inc., Minneapolis, Minn.

BORG, E. H., Member of Firm, Proudfoot-Rawson-Brooks & Borg, Des Moines, Iowa

BROWN, J. S., Sales Engr., Frigidaire Corp., Chicago, Ill.

CAMPBELL, BOWEN, Engr., Campbell Htg. Co., Des Moines, Iowa

CHARLES, P. L., Mgr., Walsh & Charles, Winnipeg, Man., Can.

CHERNE, R. E., Engr., Carrier Corp., Syracuse, N. Y. (*Advancement*)

CHRONE, R. E., Engr. & Chief Designer, R. F. Taylor, Houston, Texas

CLOSE, ROBERT, Chief Air Cond. Engr., N. B. C., Radio City, New York, N. Y.

COUGHLIN, R. J., Sales Engr., Gifford-Coughlin Co., Royal Oak, Mich. (*Reinstatement*)

CRANE, R. S., Air Cond. Engr., Delco-Frigidaire Cond. Div., Dayton, Ohio

DAFTER, E. H., Dist. Chief Engr., Carrier Corp., Philadelphia, Pa.

ELLIOT, G. B., Sales Engr., Francis Hankin Co., Montreal, Que., Can.

FEDDERS, M. P., Development Engr., Minneapolis-Honeywell Reg. Co., Minneapolis, Minn.

FELLOWS, J. R., Assoc. in Mech. Engrg., University of Illinois, Urbana, Ill.

FOLSOM, R. A., Vice Pres., W. R. Ames Co., San Francisco, Calif.

FULLER, E. W., Sales Engr., Delco-Frigidaire Div., Dallas, Texas

GIFFORD, E. W., Chief Engr., Airtemp Construction Corp., Detroit, Mich. (*Reinstatement and Advancement*)

HAMACHER, K. F., Partner, Hamacher & Williams, Milwaukee, Wis.

HARRIS, A. M., Sales Mgr., Baker Ice Mach. Co. of Texas, Fort Worth, Texas

HARRISON, J. C., Design Engr., Texas Air Cond. Co., Fort Worth, Texas

HOPPE, M. F., Mech. Engr., Pierson & Wilson, Washington, D. C.

JOHNSON, L. O., Sales Engr., H. Y. Keeler Co., Huntington, W. Va. (*Advancement*)

KESSLER, C. F., Asst. Prof. Mech. Engrg., University of Michigan, Ann Arbor, Mich.

KIRKBRIDE, J. O., Engrg. & Sale Htg. & Vtg. Equipt., Parent & Kirkbride, Philadelphia, Pa.

KOLB, F. W., Factory Repr., American Air Filter Co., San Francisco, Calif.

KUREK, T. C., Mech. Engr., Henrici-Lowry Engrg. Co., Kansas City, Mo.

LA RUE, PERRY, Dir. of Bldgs. & Grounds, Industrial School Dist., Des Moines, Iowa

LASETER, F. L., Mgr. Central Heat Dept., Atlanta Gas Light Co., Atlanta, Ga.

LONG, H. P., Sales Engr., Buffalo Forge Co., Buffalo, N. Y.

MCKERLIE, JARDINE, Editor, Refrigeration & Air Cond., Gardenvale, Que., Can.

MICHIE, D. F., Engrg. Sales Dept., Crane, Ltd., Winnipeg, Man., Can. (*Advancement*)

MOLFINO, PHILIP, Chief Draftsman, Leland & Haley, San Francisco, Calif.

MORGAN, A. S., Mgr., Fess Oil Burners of Can., Ltd., Toronto, Ont., Can.

NEWTON, A. B., Control Systems Div. Engr., Minneapolis-Honeywell Reg. Co., Minneapolis, Minn.

NUTTING, H. G. D., Sales Engr., Vilter Mfg. Co., Detroit, Mich.

PARENT, H. M., Partner, Parent & Kirkbride, Philadelphia, Pa.

PASSUR, N. A., Air Cond. Engr., Southern Pacific Co., San Francisco, Calif.

PESTERFIELD, C. H., Instructor, Mich. State College, East Lansing, Mich. (*Advancement*)

POND, W. H., Sales Engr., Fitzgibbons Boiler Co., New York, N. Y.

POWELL, R. W., Jr., Mech. Engr., Geo. W. Powell, Jr., Philadelphia, Pa.

PREBENSEN, H. J., Vice Pres., Air Comfort Corp., Chicago, Ill.

SCHLICHTER, C. F., Dist. Mgr., Surface Combustion Corp., Washington, D. C.

SEAL, A. T., Construction Supervising Engr., Airtemp, N. Y. Co., Inc., New York, N. Y.

SHIPLEY, S. C., Engr., Minneapolis-Honeywell Reg. Co., Minneapolis, Minn.

STURDY, O. C., Sales Engr., Foster Wheeler, Ltd., Toronto, Ont., Can.